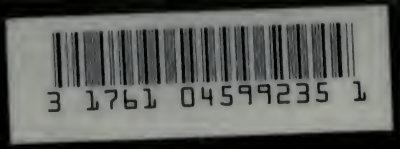
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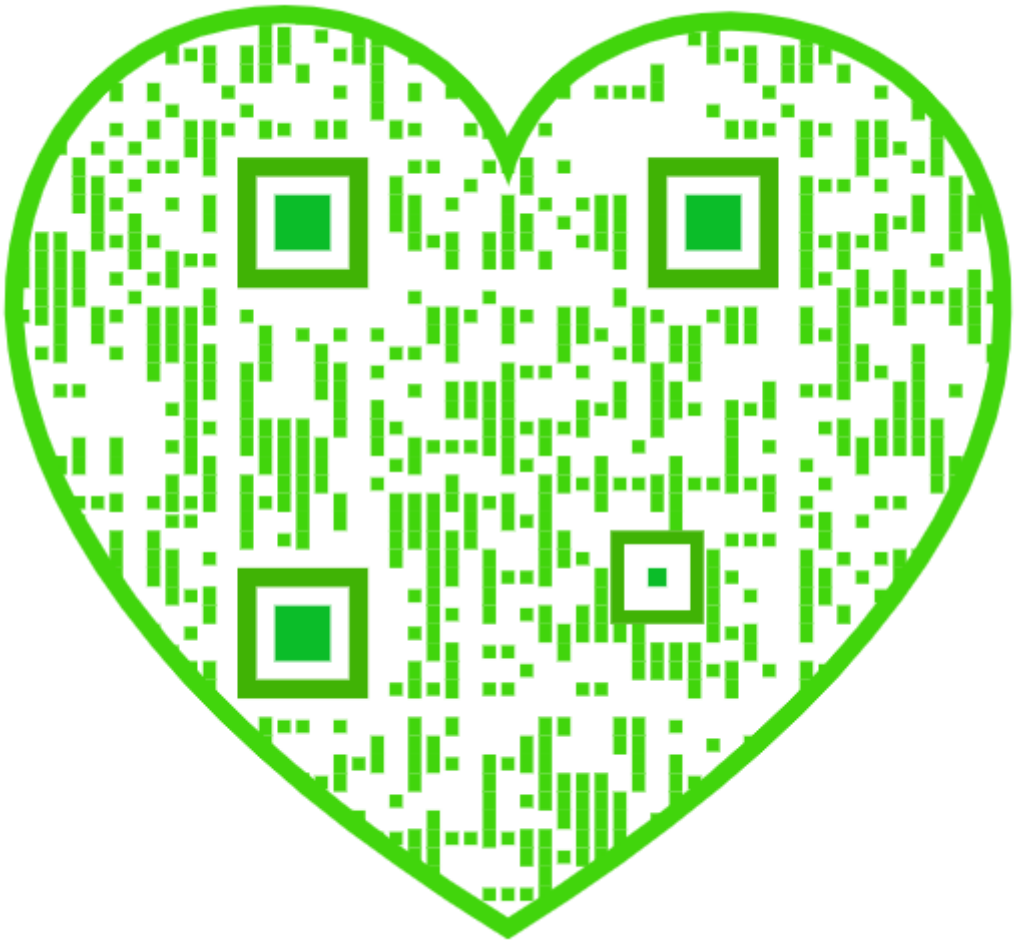

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THE PRINCIPLES
OF
MECHANICAL REFRIGERATION

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THE PRINCIPLES OF MECHANICAL REFRIGERATION

(A Study Course for Operating Engineers)

BY

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ASSOCIATE PROFESSOR OF REFRIGERATION, UNIVERSITY OF ILLINOIS. MEMBER, AMERICAN SOCIETY OF MECHANICAL ENGINEERS, AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, HONORARY MEMBER, NATIONAL ASSOCIATION OF PRACTICAL REFRIGERATING ENGINEERS. FORMERLY, ADVISORY ENGINEER—REFRIGERATION, IN THE CONSTRUCTION DIVISION OF THE WAR DEPARTMENT

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PREFACE

The Principles of Mechanical Refrigeration is based on a series of twenty articles published in *Power* during 1920. These articles were written as a "study course," the attempt being made to present the subject in as clear a manner as possible, each article being complete in itself, while the series covered the entire field of refrigeration in an elementary manner. The success of the articles seemed to warrant their collection under one cover in book form.

In compiling this book the "study course" has been rearranged, and amplified. The attempt has been made to continue the same manner of presentation; namely, to write in a simple manner, to use very little mathematics and to strive to explain the action of refrigeration by analogies to steam machinery and steam cycles. It is a book on refrigeration for practical steam and refrigerating engineers, written somewhat in the form of lectures, and with a frank expression of personal opinion in regard to designs and types of machinery and equipment. It is hoped that others besides practical engineers will find a book of this sort of interest, both in reviewing the subject, and in the use of the diagrams and tables which have been made as nearly up-to-date as possible.

It is always a pleasant privilege for an author to acknowledge assistance in the preparation of a book while still reserving all the responsibilities for short comings. The present author wishes to mention with special gratitude the assistance of Mr. B. F. Kubaugh, of the Henry Vogt Machine Co., of Mr. E. C. Lloyd of the Armstrong Cork Co., of Mr. Louis Morse of the York Manufacturing Co.,

and Mr. J. C. Goosmann, of the Automatic Carbonic Machine Co. Besides this particular thanks are due to Mr. Fred Low of *Power* for permission to reprint the articles already mentioned and to use such cuts prepared by *Power* as was deemed desirable, and to the manufacturers who have kindly supplied illustrations of their machines.

H. J. M.

University of Illinois,
Urbana, Ill.

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Principles of Mechanical Refrigeration.

CHAPTER I

Introduction

Mechanical refrigeration is not a science by itself, but is an application of heat engineering which makes use of the same laws that apply to steam engineering, heating and ventilation, air compressor work and other similar branches of mechanical engineering. In explaining the subject of refrigeration, or in making its study, it is very desirable to connect it as far as practicable with these other branches of engineering, and more particularly to understand the fundamentals underlying all of these other engineering subjects. Therefore it seems wise to review at the beginning of this discussion certain basic principles upon which matter discussed in later chapters depends.

Heat. *Heat* is a form of energy like electrical, chemical and mechanical energy and may be converted into each of these other forms. It makes itself known to us in two ways;—in the form of sensible heat and as a latent heat.

Sensible heat is that kind which can be perceived by the senses—it is evidenced by the temperature. All substances are understood to be composed of very small particles* which

*These particles are called molecules.

are in continual motion within certain limits. With a solid the limits of this motion are much more confined than in the case of a liquid or a gas. As the substance absorbs heat these particles increase in intensity of vibration and the substance becomes *hotter.* If another and colder object is brought into contact with the first one the colder substance is heated; that is, there is a tendency in the particles of the colder body to increase in intensity of vibration. At the same time there is a tendency in the hotter body to get colder, for it is giving up some of its energy to the colder one. Heat is transferred from one substance to another having a colder temperature,—never to a hotter one.

Besides the *sensible* heat, a substance may store up heat energy at a constant temperature. In changing from water to steam a large amount of heat energy is stored up in the substance as a *latent (hidden) heat of vaporization,* due to the separation of the particles of the water to the condition of a vapor. Energy is supplied in order to separate these particles against their mutual attractions, and in order to do enough *external* work so as to push aside the surrounding vapor necessary for this increase of volume. The sum of these two quantities, the work of separation of the particles of the substance and the so-called external work, is called the latent heat of vaporization, and this is always in evidence whenever a substance (as water, ammonia, carbon dioxide or ethyl chloride) boils.

In order to measure the amount of heat added to or taken from a substance, it is necessary to have a unit. For example, it would take much less heat to raise one pound of iron one degree F. than to raise one pound of water one degree. Likewise the heat capacity of salt *brine* is less than that of pure water. Therefore in order to have a reliable, standard unit of heat measurement, the British thermal unit (B.t.u., so-called) has been selected, and is defined as *the amount of heat necessary to raise one pound of water from 63 deg. F. to 64*

deg. F. The sensible heat capacity of a substance is called the *specific heat,* or the ratio of the amount of heat necessary to raise the substance in question one degree Fahrenheit to the amount necessary to raise an equal weight of water from 63 to 64 deg. F.

In the United States the Fahrenheit scale of temperature is used almost entirely by steam and refrigerating engineers with the melting point of ice at 32 degrees and the boiling point of water (at sea level and under normal atmospheric pressure) at 212 degrees. There are, therefore, 180 degrees of temperature between the melting point of ice and the boiling point of water. The Centigrade scale, with 100 degrees between these two points, is seldom used in the engine room, although it would be more handy if its use were general.

Likewise the old English units of work are used, the *foot pound,* defined as the work done in raising one pound weight a distance of one foot. A foot pound is also the work of lifting 10 pounds 1/10 of a foot or 12 pounds one inch. In each case the *work done* is equal to the product of the weight or force applied, in *pounds,* by the distance in *feet* through which the force acts. It has been found that *heat energy* is convertible into *work* and work into heat energy. The steam engine, the turbine or the oil engine convert heat into mechanical energy, and the friction of the journals in their bearings, or windage, or the action of stuffing boxes, cause the conversion of mechanical into heat energy. But it has been found that there is a certain particular relationship between these two forms of energy, and that 777.64 (or roughly 778) foot pounds of work may be converted into one B.t.u. Unfortunately it is not possible (practically) to reverse the process completely.

Properties of Vapors. All substances have three states, *solid, liquid,* and *gaseous.* The gaseous state may be saturated, superheated or a perfect gas. For instance, steam is usually saturated, or possibly slightly superheated. Air is a so-called

perfect gas at usual temperatures and pressures, but in reality it is only a highly superheated vapor. A vapor is a gas at or near the point where liquefaction occurs, and it behaves differently from a perfect gas as regards the action of heat or when work is performed on it. In order to change from a solid to a liquid or from a liquid to a vapor a certain amount of heat must be supplied in order to make it possible to separate the particles and thereby to form a new state of aggregation. As an example: it requires 143.4 B.t.u. per pound to change ice at 32 deg. F. to water at 32 deg. F.

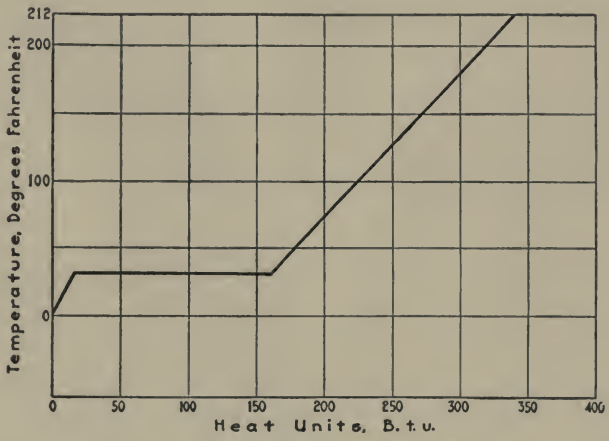


Fig. 1. The heat capacity of ice

(Fig. 1). There has been no change of temperature, but only a breaking up of the solid state into a liquid state of aggregation. Likewise water at *212 deg. F.* requires 971.7 B.t.u. per pound to change it into dry and saturated steam at *212 deg. F.* In the case of ammonia, or carbon dioxide, or in fact any liquid, there is required a certain amount of heat in order to *boil* the liquid into the state of a vapor and while the boiling pressure remains constant the temperature of boiling will not change. This heat required to boil a liquid

is called the *latent heat of vaporization* and is different in amount for the various substances with which we have to deal.

The internal energy in a substance is the amount of heat energy stored up in it. If water at 32 deg. F. is heated to 212 deg. F. there will be 180 B.t.u. stored up in each pound. This heat is stored up as so-called sensible energy because of the vibration of its particles,* and the amount of this internal energy is found by observing the temperature in the steam, ammonia or other vapor tables. The amount of internal energy thus stored up in one pound is called the *"heat of the liquid"* and it has been usual to measure it from 32 deg. F. even in the case of refrigerants which are mostly used below this temperature (in which case the heat of the liquid becomes a negative quantity).

The *total heat* of a substance is the heat necessary to raise one pound from 32 deg. F. to the condition in question. For example, the total heat of a mixture of 90 per cent steam and 10 per cent water at 212 deg. F. is

Heat of the liquid at 212 plus 0.9 times latent heat of evaporation at 212, or $180 + 0.9 \times 971.7 = 1054.5$ B.t.u.

A substance can be *superheated,* that is heat in addition to the amount required to produce dry and saturated vapor is added. The total heat of dry and saturated vapor at 212 deg. F. is

$$180 + 1.0 \times 971.7 = 1151.7 \text{ B.t.u.}$$

And if this vapor is heated still more the temperature will rise (usually with the pressure constant). The amount of heat added to attain a certain number of degrees of superheat will vary with the so-called *specific heat* at *constant pressure,* an amount very nearly 0.5 for steam and ammonia for medium pressures and small amounts of superheat.

All saturated vapors have a certain relationship between the pressure and the temperature. At standard atmospheric pressure water will always boil at 212 deg. F. and ammonia will

*i. e., the so-called molecules.

boil at minus 28.2 deg. F. Steam at 150 lbs. pressure (absolute) will always boil at 358.5 deg. F., and will *condense* at 5 lbs. absolute pressure at 162.25 deg. Likewise ammonia at 20 lbs. absolute will always boil at minus 16.65 deg. F. and will condense under 100 lbs. absolute pressure at 56 deg. F. There is, then, always some particular pressure of saturation corresponding to any particular temperature and vice versa.

The pressures dealt with in engineering have to be given in *absolute* pressures very frequently. All steam engineers know that the lowest back pressure they can obtain on their engine or turbine is that corresponding to the barometer. A perfect vacuum means a zero *pressure,* which is at sea level 14.69 lbs. per square inch below the zero of the pressure gage. Therefore, in order to change gage to absolute pressures it is necessary only to add the equivalent of the barometer pressure (usually 14.7 lbs.) to the gage pressure.

The *density* of the vapor is one of the most important factors which one has to consider. By density is understood the weight of one cubic foot of the substance, steam, ammonia or sulfur dioxide. The *specific volume,* found by dividing unity by the density, is the quantity usually used, and is the volume in cubic feet of one pound at the pressure under consideration. The increased volume of steam after expansion is the cause of the much greater exhaust openings as compared with the high pressure steam connections. Likewise the ammonia compressor is affected by the specific volume of the ammonia, because the volume of *one pound* increases very greatly with a reduced back or suction pressure.

The reader is probably puzzled to see the relationship of the preceding matter to the subject of mechanical refrigeration. In other words, why speak of "heat" in dealing with the manufacture of cold? And yet heat is a relative term, as, for instance, our winter may seem very hot to the Eskimo. Ammonia boils at 28.2 degrees below zero Fahrenheit at normal atmospheric pressure, and yet that is hot indeed com-

pared with the boiling of liquid carbon dioxide at minus 70.8 deg. F. and each of these is very hot as compared with the boiling temperature of liquid hydrogen at —422.5 deg. F. Again it is understood that it is possible to freeze liquid hydrogen at a temperature which would be colder than —422.5 deg. F. and hotter than the absolute zero of temperature where we assume no motion of the particles that compose matter. Thus compared with these low temperatures, mechanical refrigeration—dealing as it does with temperatures around zero degrees Fahrenheit, and seldom below minus 30 degrees,—is really hot, although engineers feel that it is cold enough as compared with the temperature of boiler steam or even the engine room temperature.

However, what is this process of mechanical refrigeration, and how is it accomplished? Refrigeration, in fact, is an old acquaintance, one that we meet face to face every day in one or another of its forms. The wind from the snow-capped mountains, or from a floe of icebergs, or from a glacial pond or lake, gives a sensation of cold. By this we mean that the wind is colder than the atmosphere had been previous to its coming, and it can be said that we are cooled by air *convection*, the air being previously chilled by being brought into actual contact with melting snow or ice. The street watering wagons, however, passing down the hot streets and wetting the sizzling pavements give us a sensation of relief and, for a time at least, a pleasant cool feeling in a manner similar to the effect of a summer rain.

The action of the sprinkling wagons and that of the rain is different from that of the cold wind. The former is an actual case of *vaporization* of water at a certain temperature which is always *below* that of the atmosphere. In fact evaporation is at the so-called "dew point"* temperature which

*The "dew point" is the condition when the air is saturated with water, and a lowering of temperature will cause some liquefaction of water to occur.

in turn depends on the humidity of the atmosphere. The evaporation action is similar in process and effect to that taking place in cooling towers and spray ponds for in each of these cases some *vaporization* of the water takes place—at the "dew point" temperature—which being colder than the surrounding atmosphere makes a lower resulting temperature. Again there is the action of ether or other volatile liquid on a person's palm. The ether evaporates and leaves the hand decidedly cold. These are all examples of refrigeration, there being developed in each case an atmosphere colder than there had been before the process began. But what connection have these instances with *mechanical* refrigeration and how is the latter obtained? Mechanical refrigeration is simply the term used to denote the generation artificially of a colder temperature than that of the atmosphere. Where this colder temperature is obtained by human efforts it can be said to be an illustration of mechanical refrigeration. There is no requirement in regard to temperature except that it must be below that of the atmosphere. It may be 33 deg. F. or a sharp freezer at —15 deg. F. or it might be the cooling of air some 10 to 15 degrees below that of the atmosphere.

Cooling with Ice. In the development of refrigeration, before other and better means were found, the use of ice was a successful means to an end, and of course is the best means at the present time for small refrigerators. The principle underlying the use of ice is simply the fact that ice melts at 32 deg. F. and cools the surrounding air or commodities to very nearly its own temperature. It can do this only by absorbing heat (at the rate of 144 B.t.u. per pound of ice) from some substance at a *higher temperature.* The surrounding substance which may be air, water, a metal container, or solids placed around or about it, is hotter than 32 degrees and gives up enough heat to cause a certain number of pounds of ice to melt. The fact that it requires 144 B.t.u. per pound of ice to melt means that considerable cooling can be obtained with

a very small *weight* of ice. But temperatures below 32 degrees cannot be obtained with ice alone, and as a rule the temperature of the material cooled is considerably greater than 32 degrees. It follows then that ice alone cannot be used

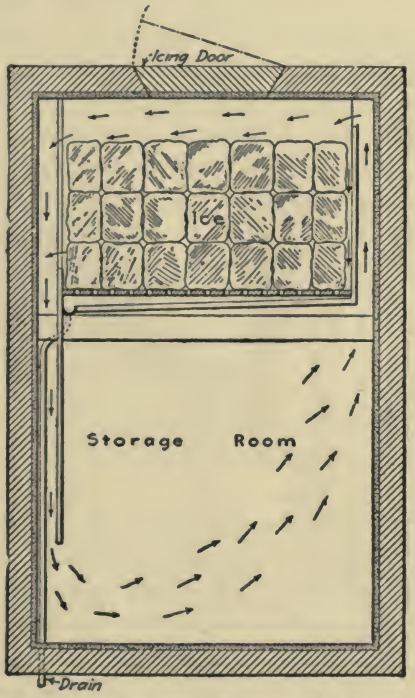


Fig. 2. A typical section of an ice bunker

for cold storage work at 25 degrees, or for any condition requiring a temperature below the temperature of melting ice.*

In the *ice bunker* (Fig. 2) heat from the *goods* being stored or heat from the outside of the storage room, or from warm air entering during the opening of the doors causes the air

*Of course ice and salt or ice mixtures containing certain acids or chemicals will give much lower temperatures than 32 deg. F.

in the lower part to increase in temperature. If air circulation is possible there will be a flow of air (as shown by the figure) due to the difference in density of air near the ice and of air below the ice loft where the eatables are placed. By difference of density is meant that there is an actual difference in the *weight* of one cubic foot of air in the two different portions of the storage room. The circulation of air is obtained in the same manner as the draft up to the stack of a power plant. As a result of the proper arrangement of air passages air will pass from the lower part to the loft where it gives up some heat, and be cooled in the process on account of melting some ice, and, by reason of its greater density, flow to the lower part again for a second round of the cycle. In other words, the air in the room is a "carrier" of heat and if the box is designed properly with proper sized air passages so that a good circulation is provided, and sufficient surface is arranged for so that the air can cool itself satisfactorily, then the box will never get too warm but will preserve the meats and vegetables from rapid decay.

On analyzing the foregoing, it will be seen that a temperature below that of the atmosphere may be obtained by securing a substance which can absorb heat at or below the temperature required in the cold room. In addition there must be a carrier of cold (and heat) that will take the surplus heat in the goods, and the heat leakages from the walls, floor and ceiling and convey it to the refrigerator (the ice) where the surplus heat will be delivered and the air, relieved of its heat, returned to the lower part again. It is apparent that success may be achieved only by having sufficient refrigerator surface, sufficient ice to absorb the heat to be removed, and proper air passages so that an unrestricted flow of air is possible. The process of heating by means of hot air is very similar, for air is heated in a furnace or bunker coil and then passed through ducts by natural or forced draft to the registers in the several rooms. As the hot air comes into contact with cold walls,

windows, etc., it gives up part of its heat and (because of its greater density) flows back to the furnace or heater again. In both cases air is a carrier and stores up *heat* as internal energy produced by the rise of temperature. It can be shown readily, however, that the carrying capacity of air is very small, and that large volumes must be provided in order to get results.

Cooling by Means of a Volatile Liquid. The objections to the use of an ice box can be seen very readily except in the case of household or small boxes which require only nominal cooling temperatures, because of the expense, dirt, icing troubles and limited cooling temperature. It is readily perceived that some means must be used that will give any desired temperature at as little cost of operation as possible. Ice and salt can be used for temperatures below 32 degrees but no engineer would consider such an arrangement for any except very special conditions. Therefore, some other means has to be found, one that will be of nominal first and operating cost.

As already mentioned there is another method of securing artificial cooling,—one using a liquid that will evaporate readily at a temperature below that desired in the room to be cooled. This action is similar to that which takes place on sponging a fevered patient with alcohol, applying ether to one's hand or water to a hot pavement, for in each case heat is absorbed from its surroundings and boils. The boiling is at a temperature dependent on the so-called vapor pressure for that particular liquid. As the liquid changes into a vapor it "carries" away from the surrounding substances an amount of heat equal to the heat of vaporization of the liquid. These substances become colder, or are maintained at a given cold temperature subject to the conditions which prevail. (Fig. 3.)

In order to grasp thoroughly what is taking place under these circumstances it will be profitable to see what takes

place in the case of steam, remembering that all vapors behave in a similar manner, and that the phenomena are identical in each case, subject to special physical properties of the given media. For example, we know that steam boiling at 150 lbs. gage has a temperature of 365.9 deg. F. but that at sea level the temperature is 212 degrees with atmospheric pressure. Likewise, with a barometer pressure of 29.92 in. of mercury the temperature of steam at 26 in. vacuum in a surface condenser is 125 deg. F. and at 29 in. is 76 deg. F. There-

Fig. 3. A test tube containing ammonia.

fore, both when *boiling* or *condensing,* steam has a temperature corresponding to a particular pressure and only one such temperature, thereby obeying the law of pressure temperature relations of saturated steam. These values are determined by careful experiment, and may be found by referring to the proper tables. Sometimes, however, the vapor becomes *superheated,* in which case the temperature has no relationship to the pressure. But as regards refrigeration, and the production of a depressed temperature, it is evident that water vapor

cannot be used very readily and that something else will have to be substituted. By process of elimination this something else has been found to be *ammonia*,* because of nominal pressures and high heat capacities.

The operating engineer feels at home with pressures similar to those he is used to and ammonia and steam are practically identical in the range usually employed. For example, ammonia has usually from 100 lbs. to 200 lbs. on the high pressure side and from zero to 20 lbs. gage on the low pressure side. These are comparable with steam engines running at high pressure. With ammonia at 15.72 lbs. gage the boiling

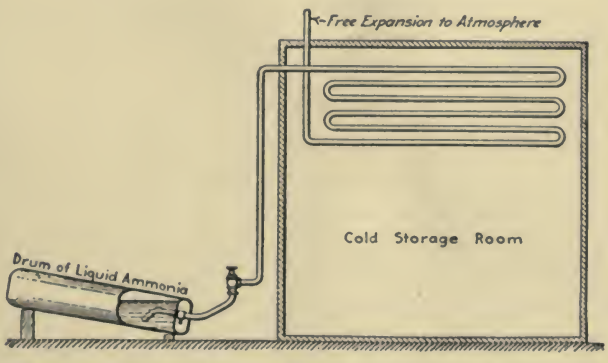


Fig. 4. The simplest refrigerating plant,—a drum of ammonia with free exhaust to the atmosphere

temperature is zero degrees F. and at 20 lbs. gage it is 5.52 deg. F. but with zero gage pressure the temperature is minus 28.2 deg. F.

Besides this nominal pressure to be dealt with there is also the matter of the storage capacity for heat. The meaning of the storage capacity of a substance may be brought out by

*Chapter VII explains why ammonia is so popular in the United States. On account of this popularity ammonia is referred to in many cases where the word "refrigerant" could be used to equal advantage.

an example. If air is used there would be required a change of temperature of 55 deg. F. per cubic foot (or 55 cu. ft. with a rise of one degree) to get the heat capacity of one heat unit (1 B.t.u.). The difference is very marked with ammonia. One cubic foot of ammonia has stored up in it as latent heat of vaporization some 20,000 heat units which would be available without change of temperature during the boiling process. Being a noxious gas the ammonia must be made to boil *inside* of suitable coils (Fig. 4) so that first the coils are cooled and then by the absorption of heat by the ammonia the air is cooled also. In the most elementary cycle the ammonia after being vaporized is allowed to pass into the atmosphere and a new supply of liquid ammonia is drawn from the drum as shown. Such a scheme, however, though it would work perfectly, would be very expensive. The cost of a ton of refrigeration per 24 hours (288,000 B.t.u.) would be approximately $200 and some 600 lbs. of ammonia would be required to do the job. Such an arrangement would be inconceivable, both from the standpoint of cost and the annoyance resulting from the odor. What then can be done?

The Ammonia Pump. If the vaporized ammonia is not to be thrown away then it must be removed from the coils promptly after vaporization and must be returned again to the condition of a liquid. This must be done continuously without interfering with the steady boiling of other ammonia in the coils, and it must be liquefied promptly enough so that it may be returned to the cooling coils for a second round of the cycle. But how can this be done? The answer is,—by the use of a machine, a gas pump which will remove as many cubic feet of ammonia vapor per minute as are boiled into vapor in that time. Besides this the ammonia pump (the compressor) will have to compress the vapor to some higher pressure, one high enough to enable the vapor to condense again in some condensing equipment. First let us consider

the principle involved in this cycle and see whether we have an old friend disguised in a new make-up.

The steam engine takes steam which had been made in a steam boiler by first heating up the boiler feed water and then vaporizing this water. The steam thus generated at, say, 100 lbs. gage pressure absorbed heat from the flue gases resulting from the combustion of fuel on the grates and in the combustion chamber. When this steam entered the steam engine cylinder it did *work* on the piston, expanding during the process, and finally was exhausted into the steam condenser under greatly reduced pressure where cooling water or some other *refrigerating* agent took away the heat of vaporization at the condenser pressure and left a liquid (water) at a temperature at or near the temperature of liquefaction. Let us repeat the cycle. Water is heated until a temperature is reached where evaporation begins, then evaporation continues at constant temperature (and constant pressure) until all the water is steam, then *work* is done *on* the piston of the steam engine, accompanied by a drop of pressure and temperature and finally the low pressure steam is condensed in a water-cooled condenser or refrigerator. Note that condensation and evaporation were separated by *work done* on the steam engine. In refrigeration exactly the same things take place, but in the reverse order.

Suppose that it is required to boil the ammonia at a temperature of zero degrees F. The resulting ammonia vapor then passes off at a low temperature,—70 degrees (perhaps) below the temperature of the atmosphere. In order to liquefy the ammonia again it must be made to give up the heat it contains to the atmosphere (which is 70 deg. F. in this case). As "heat cannot be made to run up hill" it is clear that something must be done so that cooling water at 70 degrees will condense this ammonia gas. The answer is to *do work on* the ammonia so that the pressure will increase to a point where 70 deg. F. water will condense it. Perhaps this is not clear to

the reader at first, unless the pressure-temperature relation is plainly understood. A few of these relations are as follows:

Pressure, lb. gage.	Temperature, deg. F.
75	50.3
100	63.5
125	74.7
150	84.4
175	93.0
200	100.9

Suppose the ammonia vapor was compressed to 200 lbs. gage, then it would *condense* at 100.9 deg. F. A condenser supplied with water at 90 deg. F. would probably satisfy this condition, whereas the same vapor at a pressure of 150 lbs. gage would condense at 84.4 deg. F. and would require water of 75 deg. F. or lower. The function of the ammonia gas pump (or compressor) is only that the gas be compressed to such a pressure as will allow the cooling water available at the plant to condense it. Liquid ammonia at 100 lbs. pressure will boil at 63.5 deg. F. and gaseous ammonia (at this pressure) will condense at 63.5 deg. F. In each case heat is made to exchange *downwards* in the scale of temperature. The compressor is simply a gas pump, one that will lift the ammonia, or other refrigerant, to a pressure level so high that the temperature of liquefaction will be above the temperature of the atmosphere (or of the condensing water at the plant). During refrigeration a certain number of heat units were abstracted from the air in the cooling rooms, or from the commodities, in order to *boil* the ammonia at the required temperature—say zero degrees—and after compressing the gas a number of heat units equivalent to the refrigerating effect *plus* the heat equivalent of the compressor work were removed by the condensing water at atmospheric pressure. In fact the condenser has to remove about 30 per cent more heat units than are obtained in useful refrigeration. The gain in so doing is that the work may be performed at the temperature of the river or pond or well, irrespective of

whether the refrigerant boiled at 30 degrees above or below zero Fahrenheit.

The Ammonia Compression Cycle. The attempt up to the present has been to show that refrigeration is obtained in practice by boiling the ammonia, and that the only use of the compressor is to make it possible to continue the use of the same charge of the refrigerant. It is very important that this reason for the use of the compressor be clearly understood. Also what it accomplishes and what is expected of it. But really there is nothing unusual about the compressor. The dry-air vacuum pump draws the rarefied air (and steam) from the condenser, compresses it to an amount slightly greater than that of the atmosphere, and then discharges the mixture into the atmosphere. The deep well pump takes water at the level of the water in the well and lifts it up to the tank, cistern or irrigation ditch. In each of these cases, namely the deep well pump, the air pump and the ammonia compressor, work is *done on* the substance pumped. Each substance is lifted in level. This level is datum level, pressure level and temperature level respectively. In the case of the compressor something has been done so as to make it possible to condense the ammonia at a much higher temperature than that at which refrigeration took place, and therefore it was a "temperature" pump. The compressor, then, should be designed, as nearly as is practically possible, to draw into the cylinder at each stroke an amount of gas equal to the piston displacement, and discharge all of this amount into another part of the system with the least expenditure of power and at the pressure required for condensation. This pressure will vary usually from 100 lbs. to 200 lbs. gage pressure, depending on the condensing water available for that purpose at the plant.

As in air compression, a kind of work which it very closely resembles, the gas from the ammonia compressor becomes quite *hot* during compression, especially if the ratio of the suction

to the discharge pressure is appreciable. In fact it is usual to find with pressures of 125 to 185 lbs. per sq. in. a discharge temperature of from 150 to 250 deg. F. which represents *superheats* of from 75 to 175 deg. F. From this fact it can be seen that the discharge from the compressor must be allowed to pass into an after-cooler, or something of like nature, so that the superheat, first, and the latent heat of condensation, second, may be properly removed. The after-cooler or so-called *condenser* is made in various designs so as to bring the ammonia gas into good *metal* contact with the water and thereby make the best practical use of the water. In consequence the ammonia will be cooled to an amount some 10 to 15 degrees above the cooling water temperature, and the ammonia in due course of events will become a liquid at a pressure slightly less than the discharge gas just outside the compressor. The condensed liquid is then collected into a drum or *liquid receiver* where it remains until it is needed for a second round of the cycle. The compressor, condenser and liquid receiver, being subject to the compressed gas, are often called the *high side,* or the high pressure side.

As mentioned, the liquid ammonia in the receiver is under pressure and will flow like water in consequence of this pressure through the pipe connection to the place where the cooling coils are placed. At the cold-storage room, or brine tank, there is a valve to regulate the flow of the liquid ammonia. This valve has the common name of an *expansion* valve, although no real expansion takes place but only a flow control so that the amount of liquid flowing will just equal the amount of liquid boiled into a vapor in the refrigerating coils. Since the pressure of the liquid passing through the valve is lowered in the process the valve may very properly be called a pressure reducing valve. The reason it is desired to secure a lower pressure is that a reduced temperature of the boiling ammonia in the coils is necessary. For example, it is frequently necessary to obtain such a temperature as

zero degrees F., which means that a drop of pressure from that of the condenser to 15.72 lbs. per sq. in. gage must occur. Therefore it is desirable to have only a very small area of valve opening, seldom larger than the lead in a lead pencil, from which the liquid ammonia squirts into the coils like water escaping through a sand hole in a water pipe fitting under water pressure. If the piping is properly arranged and proportioned, the liquid ammonia will begin immediately to boil in consequence of absorbing heat through the pipe coils from the air or brine or other substance which is to be reduced in temperature, and in so doing the ammonia becomes a gas. As a gas ammonia has little heat capacity and is

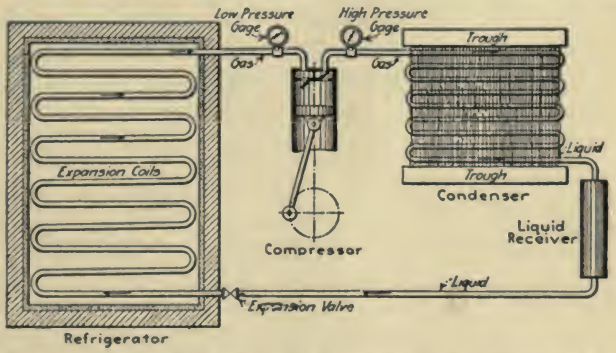


Fig. 5. A typical layout of an ammonia compression plant

therefore of little refrigerating value, hence it is essential to get rid of the ammonia gas as promptly as it is formed, and separate it from the liquid and return it through the low pressure system (the so-called suction line) to the ammonia compressor where the gas is again compressed for a second round of the refrigerating cycle.

The Refrigerating Cycle. From the foregoing it is evident that the refrigerating cycle (Fig. 5) consists of three distinct parts, the compressor, the condenser and the refrigerating coils, including the suction return. The refrigerating coils in action are much like the boiler of a steam plant. In each

case the heating surface (pipes) is in contact with relatively hot gases, the air of the room or the hot products of combustion. In the case of the ammonia pipes the hot air in the cold storage rooms boils the liquid ammonia, whether vigorously or not depending upon the temperature difference between the air in the room and the boiling temperature of the ammonia in the coils. Now in the steam boiler the steam generated must be allowed to pass out through the steam nozzle as fast as it is formed, and it must be used by the steam engine or the heating system at the same rate. If for any reason the steam ceases to be used in this manner the pressure in the boiler will rise and the safety valve will blow, thus allowing the steam to escape another way. On the other hand, if steam is used too fast, say by throwing too heavy a load on the line to be handled by the boiler, then for a time more steam will be taken from the boiler than is generated during that interval of time. The result of such conditions would be that some heat energy would be taken from the stored up heat in the hot water of the boiler, the steam pressure would drop and soon the boiler would be out of commission. In other words, in order to secure satisfactory boiler action the amount of steam generated by the boiler for each unit of time must be used during that same unit of time.

Likewise the refrigerating coils must be so ordered and arranged that the vapor boiled from the refrigerating coils will be just enough for the compressor. The compressor, being a pump, may handle a certain volume of gas per minute corresponding to a definite piston displacement for a particular machine and rate of operation. If the boiling action is too rapid and more gas is formed than can be handled by the compressor the vapor will back up in the suction line and the gas pressure will rise. If the boiling *pressure* rises then the boiling *temperature* will rise likewise perhaps to a point where satisfactory results will no longer be obtained. On

the other hand, too slow a boiling action, due to having too large or too high speed a compressor or too small a set of refrigerating coils, will result in having a drop of suction pressure and perhaps too low a *boiling* temperature of the ammonia. But the result of too low an ammonia pressure is that the volume of one pound (the specific volume) becomes excessive and the necessary *weight* of ammonia will not be pumped. Of course it is the weight of ammonia compressed by the compressor and condensed in the condenser per unit of time which is of primary importance. Therefore if the *weight* of ammonia discharged by the condenser decreases then the capacity of the machine, and also the whole plant, decreases in proportion. For satisfactory results the piston displacement of the compressor and the volume of gas boiled from the coils must be inter-related.

For example, given a room to be held at 36 deg. F. and 500 sq. ft. of cooling coils supplied, to be used normally with a 37.9 lbs. gage suction pressure which corresponds to a temperature of 24 deg. F. Then with these conditions there will be $36 - 24 = 12$ degrees difference of temperature on the two sides of the pipes, and about 5.45 cu. ft. of gas will be boiled out of the coils per minute. But how about this weight of a gas or vapor?

Perhaps it is not perfectly clear to everyone that a gas or a vapor can have weight. Yet a steam engine or a turbine operates with steam which certainly had weight as *water* previous to vaporization, and exhaust steam may be condensed in a condenser and the condensate weighed. An engine is often spoken of as having a water rate of 15, 20, or 25 lbs. of steam per horse power hour. Again there are examples of balloons and other *lighter than air airships*, which of course are able to remain in the air only because they displace a greater *weight* of air than they weigh themselves. So every pound of ammonia evaporated in the coils will

occupy a certain particular volume, which is itself dependent on the pressure. For instance, a pound of ammonia at—

200	lbs.	gage	will	occupy	1.40	cubic	feet.
150	"	"	"	"	1.82	"	"
100	"	"	"	"	2.57	"	"
30	"	"	"	"	6.3	"	"
20	"	"	"	"	8.0	"	"
10	"	"	"	"	11.0	"	"
0	"	"	"	"	18.0	"	"

In consequence, should it be desired to compress one pound of ammonia from zero pounds gage to 200 lbs. gage the volume of gas drawn in during the suction stroke would be 18.0 cu. ft., and the discharge volume (if the gas is not superheated) would be 1.40 cu. ft.

The Refrigerating Coils. The refrigerating coils have already been compared with the heating surfaces of a boiler, as in each case a liquid is evaporated into a vapor thereby absorbing heat and cooling the source of this heat (the flue gases in the case of the boiler). Taking the steam boiler as an example it will be remembered that the usual figure taken is 10 sq. ft. of heating surface per boiler horse power, and a boiler horse power as the evaporation of 34½ lbs. of water from a temperature of 212 deg. F. to dry and saturated steam at 212 deg. F. In refrigeration it is not possible to proceed in this manner, as the conditions of operation are widely varying. The boiler plant has about the same temperature of flue gas, or combustion chamber, and about the same average temperature of the water inside the boiler. Although the refrigerating coils are really a boiler yet neither the condition outside nor inside the coils is constant. Sometimes the difference in temperature on the two sides of the pipe may be 10 deg., 15 deg., or even 20 deg. F. Evidently more heat will pass through a material if the temperature difference on the two sides of the material is great, than if it is small. In other words, the greater the temperature difference between the ammonia and the outside of the pipe

the greater each square foot of surface will increase the capacity of the coils. But besides the area of exposed surface there is a very important consideration of the *condition* of the surface of the pipe. For example, the pipe surface may be wet or dry, it may be frosted slightly, or it may be very heavily frosted and covered with ice. The inside of the pipe may be covered with scale; or with a mixture of oil and dirt. It is not reasonable to expect the same amount of boiling action in all of these cases. In fact, clean pipe will always give the best results, and the kind and amount of scale, oil or other foreign material as well as other factors affect the ability of heat to pass through the pipe. There is, then, a coefficient of heat transmission, so-called, for refrigerating piping in the same manner as for steam condensers, evaporators and boilers.

As the matter of heat transfer in refrigeration is a matter which is coming up all the time it is very desirable to have some means of calculation for the different cases. This equation may be worded as follows:

The heat transmission in B.t.u. *equals* the area of the surface in square feet *times* the coefficient of heat transfer *times* the temperature difference in degrees Fahrenheit on the two sides of the surface. The coefficient of heat transfer (k) has to be found out by experience or by a test for the specific purpose of finding out the value desired. In a steam condenser this value may be 400, bleeder type of ammonia atmospheric condenser may be 150, and the ordinary expansion refrigerating coil may be 2.0 B.t.u. per *square foot* per *degree difference* of temperature per *hour*. As an example, suppose it is desired to calculate the refrigerating effect of 1,200 linear feet of 1¼-inch pipe with a suction pressure of 25 lbs. gage (corresponding to a boiling temperature of 11.3 deg. F.) and a room temperature of 34 deg. F. Although it is not possible to predetermine the amount of frost likely

to accumulate on the pipe yet the value of 2.0 for k is reasonable. Then: — the total B.t.u. transferred $= \frac{1200}{2.3} \times 2 \times (34 - 11.3) = 23{,}700$ B.t.u. per hour where the value of 2.3 is the number of linear feet of 1¼-inch pipe per square foot of outside area. It is not convenient to use the B.t.u. as the unit of refrigeration any more than it is in the case of steam boilers. In boilers a horse power is the evaporation of 34½ lbs. of water at 212 degrees to dry and saturated steam at 212 deg. F. or 34½ × 971.7 or 33,524 B.t.u. per hour. In a like manner the *ton of refrigeration* has been taken as the unit of refrigeration, and is defined as the rate of *cooling* of 12,000 B.t.u. per hour or 200 B.t.u. per minute. So, in the above problem the tonnage becomes $\frac{23{,}700}{12{,}000} = 1.97$ tons of refrigeration. A refrigerating coil for the conditions as stated, therefore, would require a compressor which would compress and deliver to the condenser enough gas to supply—when condensed—1.97 tons of refrigeration and the high and low side would then be balanced satisfactorily.

From the foregoing it will be seen, therefore, that mechanical refrigeration is heat engineering of a sort that is similar in many ways to steam engineering. The cycle has to be a closed one, but otherwise it is really similar, though reversed in the order of the events, to the more common steam engine cycle. The study of refrigeration, therefore, is more or less a study of the details of construction suitable for the refrigerant used (*e. g.*, the compressor, the condenser, fittings and accessories, erection and the methods of insulation), of the refrigerants, and of the more common uses of refrigeration. These matters will be taken up in detail in the following chapters.

CHAPTER II

The Compressor

In the foregoing chapter the refrigerating cycle was described and the function of the compressor was touched on. The compressor is considered a temperature pump, and is provided in order that the gas from the refrigerating coils may be put into such a condition that it can be liquefied again. This is the only place in the cycle where power is used, and therefore (for a certain set of operating conditions) the only place to be examined when economy of operation is in question.* (See Fig. 6.)

Also, as a pump, the compressor is the means of "lifting" the gas from the condition of low to that of high pressure. The volume of gas compressed is dependent absolutely on the compressor piston displacement. The amount of gas condensed in the condenser and subsequently boiled in the refrigerating coils depends on this piston displacement. Therefore, other things being equal, the capacity of the compressor depends on the volume of gas drawn into the cylinder per unit of time (per minute) and the degree of perfection with which this gas is compressed. This "degree of perfection" requires some explanation.

The *ammonia compressor* in particular is similar to the ordinary air compressor, except for the physical characteristics of the substances being compressed. For instance, am-

*It will be shown that the condition of the condenser and the low pressure side of the system have a decided effect on the power input per unit of refrigeration.

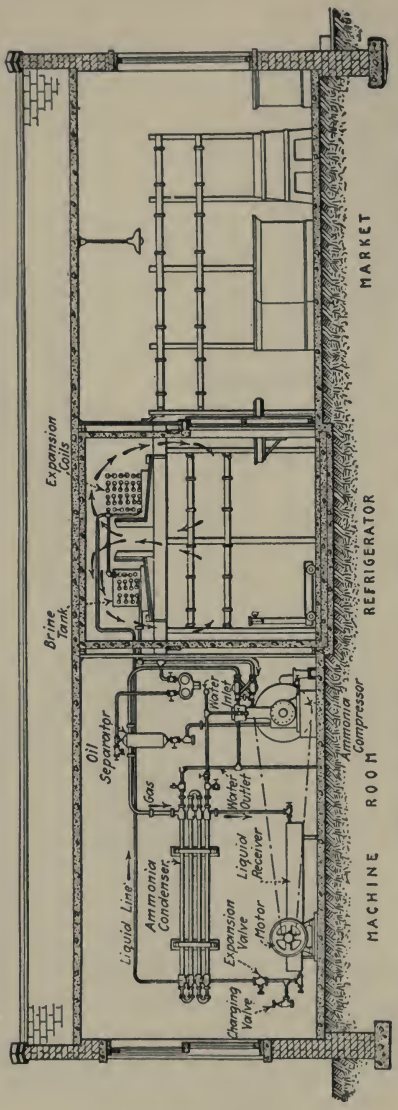


Fig. 6. A typical section of a compressor room, cold storage room and market

monia is difficult to keep confined, and being expensive, as well as obnoxious to the senses, the usual design of air compressor has to be modified:—in the density of cast iron used, the type of fittings, kinds of joints, and stuffing boxes, and forms of valves, in order to secure tightness. Although ammonia compressors work between the limits of 20 lbs. gage

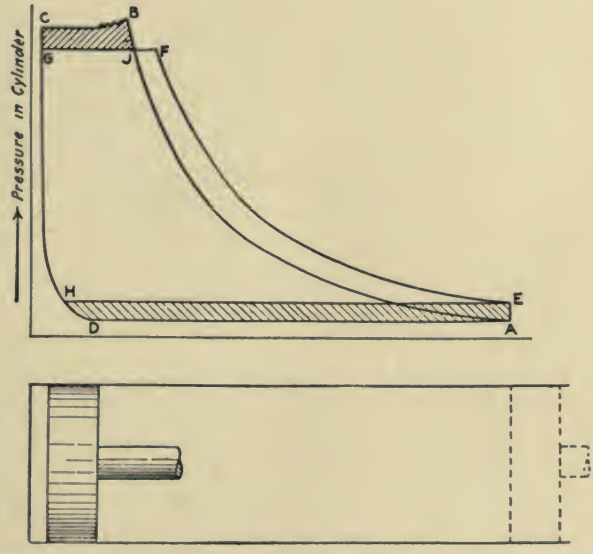


Fig. 7. Indicator card and ammonia cylinder. (The hatched area indicates excess work done in the cylinder on account of faulty design or operation

and about 160 lbs. gage under standard conditions—and frequently with even greater limits—little use is made of compound compression. In air compression it has been found economically necessary to use a low and high pressure cylinder when the discharge pressure is 70 lbs. or greater, in as much as losses during compression assume large proportions at that time. However, two stage compression is now being

used for low temperature refrigeration to a certain extent and without question a more or less general application of two stages is sure to come in the near future. The mechanically operated valve, which is commonly used in some makes of air compressors, is never used in refrigeration, and the reasons are principally those already considered. In order to operate a mechanical valve some valve rod and stuffing box would have to be made to enter the compression cylinder. This would thereby make possible ammonia leaks, as well as operating troubles in maintaining the stuffing boxes, and the impossibility of control of the compressor pressures. In general, however, it may be said that the problems in air and ammonia compression are similar and that compressor designs for both are parallel. To understand the problem during compression, it is desirable to study the action of the valves as shown by the indicator diagram in Fig. 7.

The Indicator Diagram. The indicator diagram shows the variation of the pressure in the cylinder for every position of the piston, and about all that can be hoped for in such a diagram is to be able to observe the action of the valves, though it is also a ready means of finding out whether the valves and the piston are tight.

As has been mentioned, the compressor is the pump lifting the ammonia vapor from the boiling pressure to the condenser pressure. As a pump it is desirable to reduce all losses to the minimum. Therefore, during the suction stroke, it is essential that the compressor be filled with gas at the *pressure* in the refrigerating coils, and also, that the compressed gas be discharged at the pressure in the condenser. Also, it is important to draw into the cylinder at each stroke of the piston a volume of vapor equal to the total piston displacement, and to compress this gas along a compression curve which will require the least expenditure of power. These specifications are impossible of perfect fulfilment but they are the goal we attempt to achieve.

During the *suction stroke,* so-called, gas is drawn into the cylinder of the compressor, and between successive strokes an equal volume of additional gas is evolved by the boiling action in the refrigerating coils. If the piston, Fig. 7, was working in a cylinder of zero clearance and no gas was admitted into the cylinder a perfect vacuum would be obtained with a movement of the piston. In the actual case, as the piston moves from the end of the stroke, it *tends* to create a perfect vacuum, but as soon as a certain drop of pressure *inside* the cylinder is obtained, the gas in the suction bends and suction mains commences to flow into the cylinder through the suction valves. The amount of pressure drop depends on the local conditions; as for example, the speed of the piston, the amount of suction valve area in proportion to the area of the piston, the ease of opening of the suction valve, and the size of the pipes and ports leading up to the cylinder. This drop of pressure is similar to the drop of pressure in an electric circuit, or in the case of water flowing through pipes. Certain losses of pressure are due to *velocity head,* to friction, and to impact on changing direction of flow. Therefore it is desirable to overcome these losses to as great an extent as possible by making straight connections of liberal area and by using a large valve that can be readily operated.

The reasons for care in obtaining full suction pressure inside the compressor cylinder may be seen on considering this phase of the problem. Fig. 7 shows two suction pressures, HE and DA, and the cross-hatched area shows, in part, the amount of extra work which must be used to compress the gas at these two pressures. But more important still is the fact, that the higher pressure represents a greater *weight* of ammonia. For example, if one suction pressure is at 5 lbs. gage and the other at zero pounds gage there would be about 30 per cent more ammonia by weight at the higher pressure for an equal *volume* in each case. Thus with less work of compression in the lower part of the indicator dia-

gram, and also the compression of a much greater weight, it is doubly important to reduce the drop of pressure within the compressor to the lowest amount practicable.

The *pressure of discharge* is determined by the action of the condenser, but the pressure inside the cylinder may be almost any pressure in excess of this amount. In general, the excess pressure must be great enough to overcome the friction and impact losses in getting out of the cylinder through the discharge bends and pipe line up to the condenser, and, in addition, the throttling action of the discharge valves. Sometimes, in slow acting compressors of liberal design, this excess pressure shown as BJGC in the figure, is nominal, but occasionally the additional work in getting the gas out of the cylinder becomes excessive, and should be corrected by removing the cause of the trouble.

The compression line has been given considerable attention on the part of compressor designers, but with indifferent success. It has been found for practical reasons that the most efficient compressor is the one using a water jacket to keep the cylinder cool during operation (rather than that using a liquid injection, as is done in wet compression) and that under the above conditions the compression curve cannot be very much controlled. In addition to the compression curve, the effect of clearance has been studied, and for some time it was considered very detrimental to have volumetric clearances of more than a fraction of one per cent. The reason for this may be seen in Fig. 7, which shows the expansion of the clearance gas along the line CHD. Evidently it will not be possible to draw a new supply of ammonia gas into the cylinder until the clearance pressure has dropped down to the suction pressure outside the cylinder, or even to a slightly lower pressure. This point is represented on the diagram by the letter D, and the effective piston displacement is reduced by an amount corresponding to the part of the stroke passed through in reaching D. The capac-

ity of the machine, therefore, will be reduced in proportion to the reduction of effective volume, but the gas expands along the line CHD and does *work* on the piston. This work is nearly equal to the work of compression of the clearance gas and therefore little thought is now given to providing small

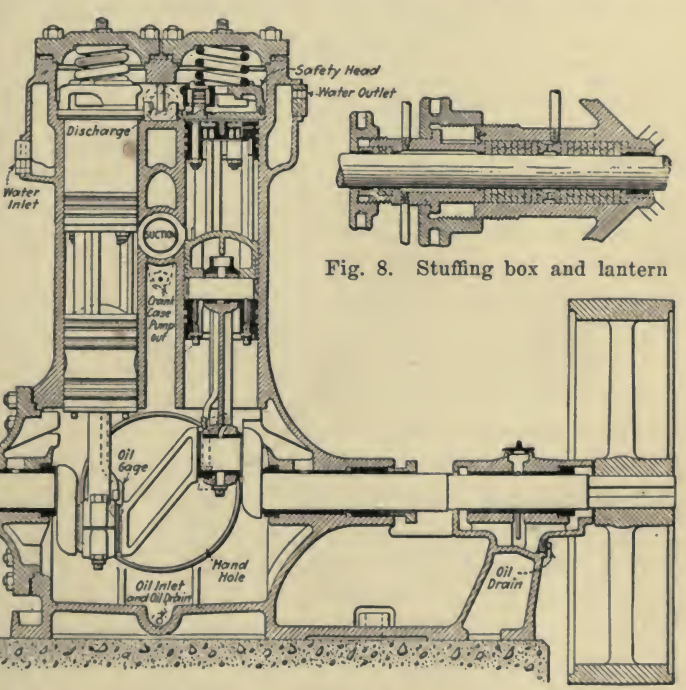


Fig. 8. Stuffing box and lantern

Fig. 9. Section of the twin vertical single acting enclosed type ammonia compressor (Vilter type)

clearances. In fact 5 to 7 per cent clearance is quite common on the modern high speed compressors.

The Suction and Discharge Valves. Probably no one part of the refrigerating system has received the careful consideration of designers more than have the compressor valves, and very justly so. The compressor represents the main cost

of operation and anything that will permit an increase in capacity or a reduction of power input per unit of refrigeration output is an important factor. In the early days designers had only the steam engine for the motive power and the compressor was direct driven at 50 to 60 r.p.m. or less. Later came higher speed engines up to 100 r.p.m. and finally the demand for direct connection with oil engines and synchronous motors, or even medium speed four valve engines. The types of valves used, as well as other compressor characteristics, have undergone a steady change. The old reliable form of poppet valve (Fig. 11) used with such excellent success at first, and still the best type for certain designs of compressors, has been found unsatisfactory with present day moderate speed machines.

The *poppet valve* is shown in detail in Fig. 12, in which it may be clearly seen that both the suction and the discharge are identical in essential features. The valve is designed for "line" contact, so that excessive unbalanced pressure will not be experienced. This would have the effect of causing a "toe" at the time the valves open and would result in unnecessary additional work in the cylinder. In addition, the poppet valve must be supplied with springs to assist in prompt closing and with some form of dash pot which will enable the valve to close without shock. With the suction valve there is, also, usually, a means of preventing the valve from falling into the cylinder should some part of the stem become broken. In Fig. 12 it is seen that this danger is lessened by means of a collar on the enlarged part of the stem.

The *plate* or the *ribbon form* of valve (Figs. 13, 14 and 15) has been growing increasingly popular for some years so that now nearly every one of the prominent manufacturers of ammonia compressors is using this valve for certain types of their compressors. At first it was feared that, because the "line" contact was not used, but instead one wherein the valve overlapped the valve seat for quite an appreciable

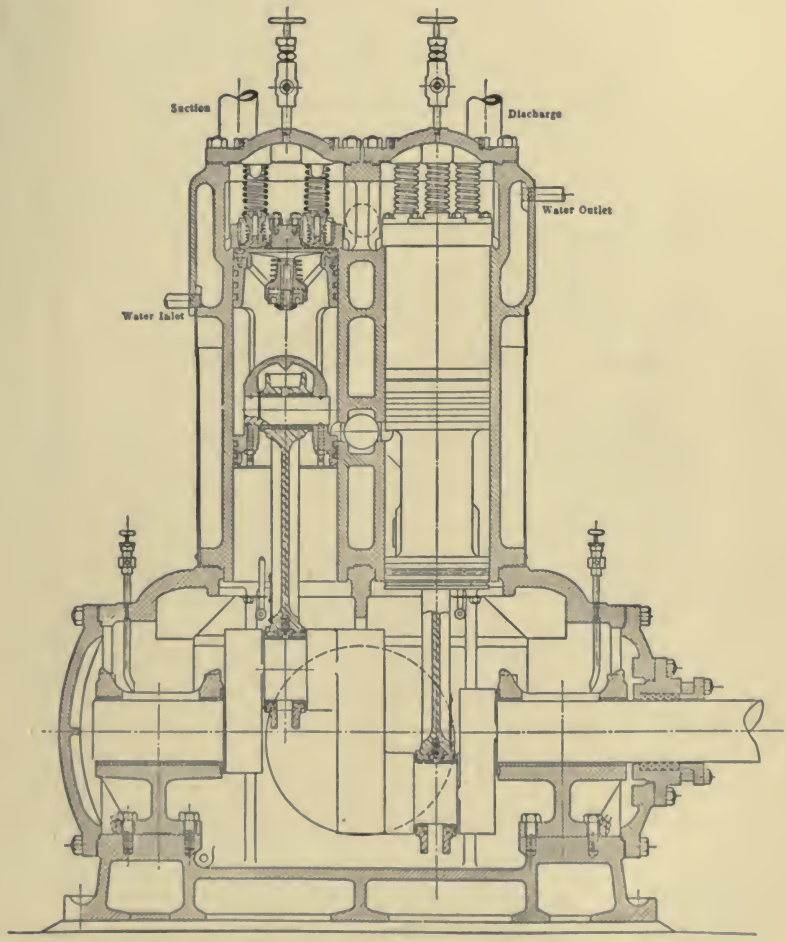


Fig. 10. The York Enclosed type vertical single acting compressor [50 to 75 tons capacity]

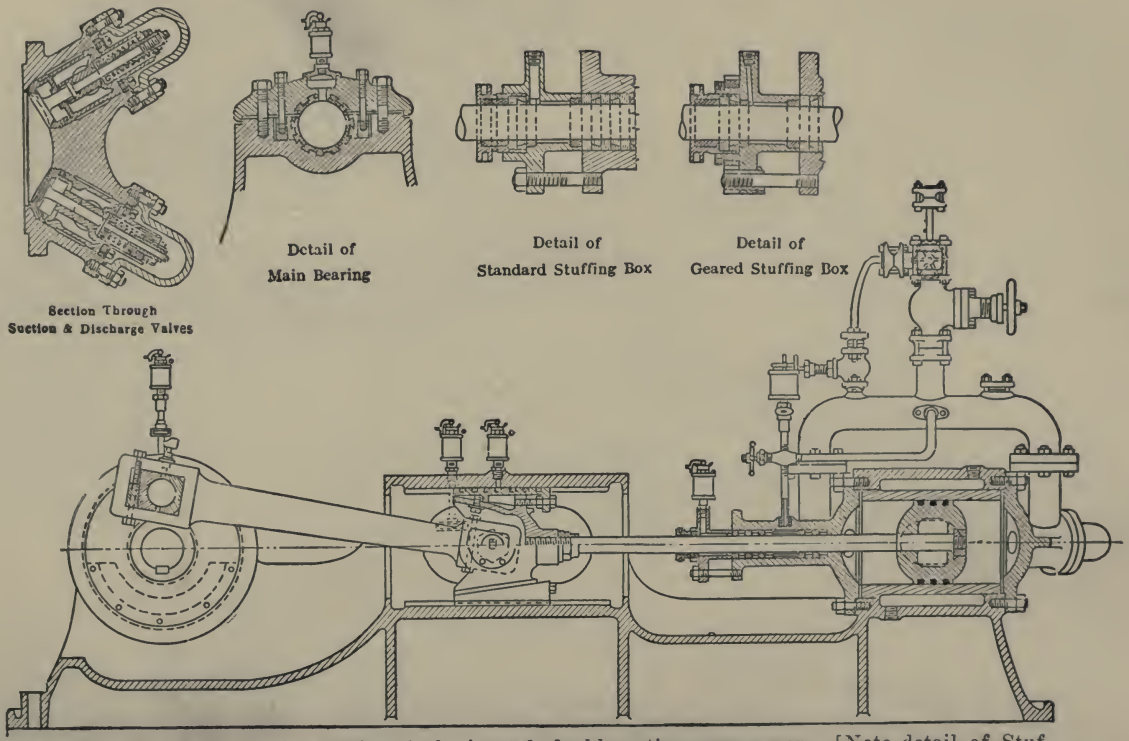


Fig. 11. Section of a Triumph horizontal double acting compressor. [Note detail of Stuffing box main bearing and crosshead]

amount, considerable (even excessive) surplus pressure would have to be used in opening the valve. Although this is true in some cases it is not serious as a rule, and frequently is not appreciable. On the other hand, advantage is obtained in the simplicity, the lack of inertia, the lack of danger from broken valves, and the reduced cost. Frequently the discharge and suction valves are identical, except for the direction of opening of the valve when assembled in place.

In the steam engine it is true that the valve gear is the most important feature of the design, but it is not the only consideration by any means, and this is true for the refrigerating compressors. In fact, many other features are of great importance and make a great difference as regards successful operation and low operating costs. For example, the main design divisions are worthy of special mention, as is also the matter of lubrication, standard operation and stage compression, high speed and multiple compression, and machines designed for other working vapors besides ammonia. A discussion of a few of these matters follows, particular attention being paid to recent features and developments.

The Compressor. The old form of twin cylinder vertical single acting compressor is still being successfully used. This design embodies the uniflow principle to a certain extent, the gas entering the lower part of the cylinder during the up stroke and passing to the upper side of the piston through a "balanced" suction valve of the poppet type in the center of the piston during the down stroke. This compressed gas is discharged through one or more poppet valves of the ordinary type, and the design is one using very small "striking" clearance—usually 1/32 to 1/64 in. A *safety head* (so-called) is provided which will lift when pressures of 250 or more pounds are experienced inside the cylinder. The balanced suction valve is one (Fig. 10) which has its weight just balanced by a spring, and which will open or close with the slightest difference of pressure on the two sides of the

piston. The valve is usually a large one, with an area of opening of about 20 to 25 per cent of the area of the cylinder. As the lower part of the cylinder is exposed to suction pressure only, the form of rod packing does not have to be very elaborate. As a rule, however, the stuffing box has from eight to fourteen packing rings with a lantern of metal approximately at the middle and is connected at the lantern with the lubricating pump. Box girder columns connect the cylinder with the frame, and the crank shaft is designed with the cranks of two ammonia cylinders 180 degrees apart,

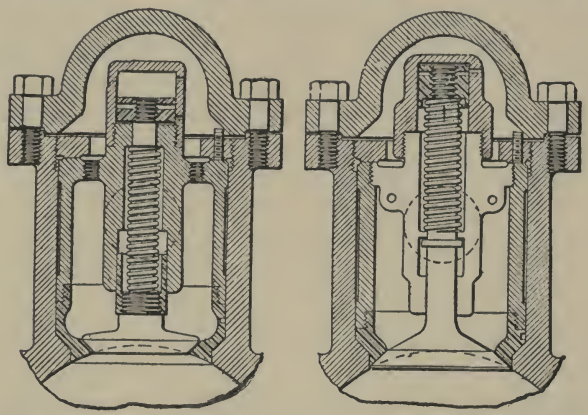


Fig. 12. Discharge and Section Valves designed for "line" contact and supplied with dash pots

and the steam engine (if one is used) connected to one of the same crank pins. A heavy fly wheel must be used also, especially for the lower speeds. (See Fig. 16.)

Smaller sizes of compressors are designed somewhat differently as shown in Fig. 8. This enclosed type, usually made up to 20 tons but sometimes in larger sizes, is generally of high rotative speeds varying with the size of the cylinder. This form of compressor is splash lubricated, having an enclosed crank case, and a rotary instead of a reciprocating rod passing through the stuffing box. The valves shown in

the figure (contrary to the design of some enclosed compressors which use the balanced suction valve) are different from the larger machine just described, having a "plate" valve in the piston and a modified plate valve with dash pot for the discharge. This discharge valve may be used in sets of two or three, and can operate without noise to over 350 r.p.m.

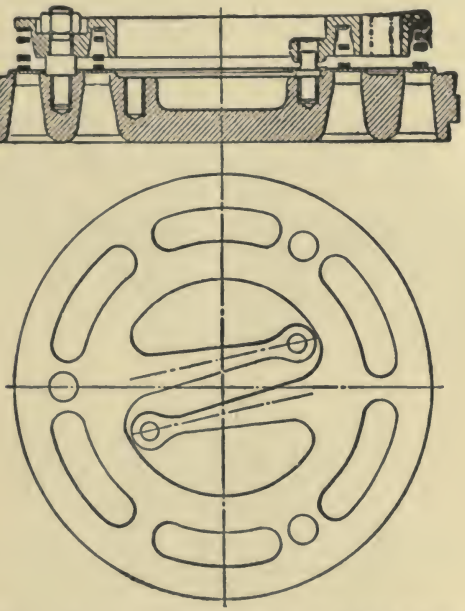


Fig. 13. The Borsig type of plate valve. [This design is subject to trouble due to possibility of breakage near the points of support]

In general it may be said that, except for the enclosed type, the *horizontal compressor* is the most popular design. It is easily accessible, and the valves are more easily removed and replaced than with the vertical types. Being double acting, practically the same tonnage can be obtained with one cylinder as with the two cylinders of the single acting design. Also the space occupied, the head room required and the

initial cost are less. As a rule the construction is heavy, in order to be capable of taking severe forces without appreciable distortion.

The general construction of the horizontal compressors is shown in Fig. 11. The frame and base is usually of close grained iron or semi-steel, the metal being well distributed in order to offer the best resistance to the severe forces which are developed during compression, and it is usually of the rolling mill or the heavy duty type of frame with spherical

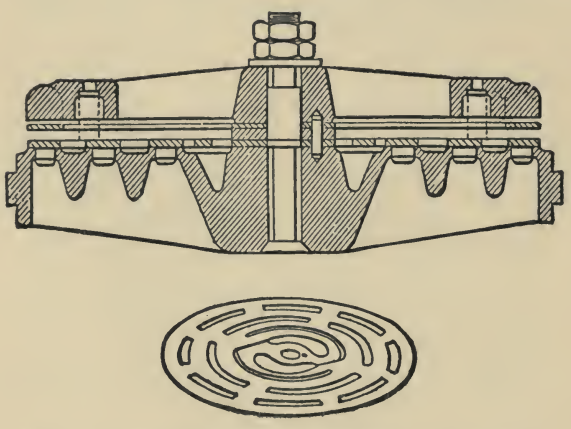


Fig. 14. The Rogler plate valve. [Note that closure occurs due to the elasticity of the plate itself]

heads. The piston is usually of semi-steel, designed for from two to four spring piston rings. As a rule the poppet type of valve only is used. The later horizontal machines designed for higher rotative speeds are even heavier in construction, or are made of special material of greater strength, on account of the increased forces developed.

Lubrication. The problems of lubrication are slightly different in the case of ammonia compression from those in the case of air. The ammonia cycle is a closed one and after

operating for some time a certain (small) amount of oil will circulate continually with the ammonia through the cycle. On the other hand, the greater part of the oil that gets past the oil trap on the discharge line from the compressor will collect in the condenser and the liquid receiver or perhaps

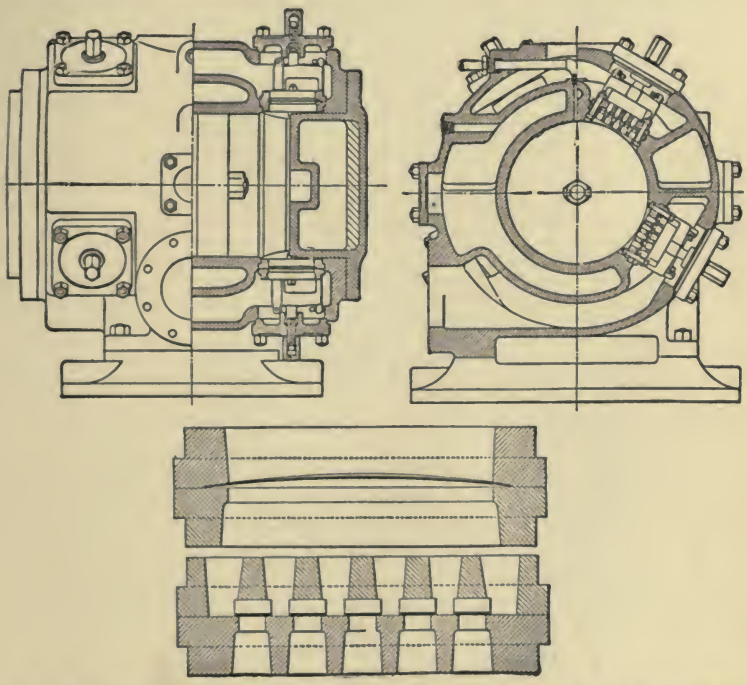
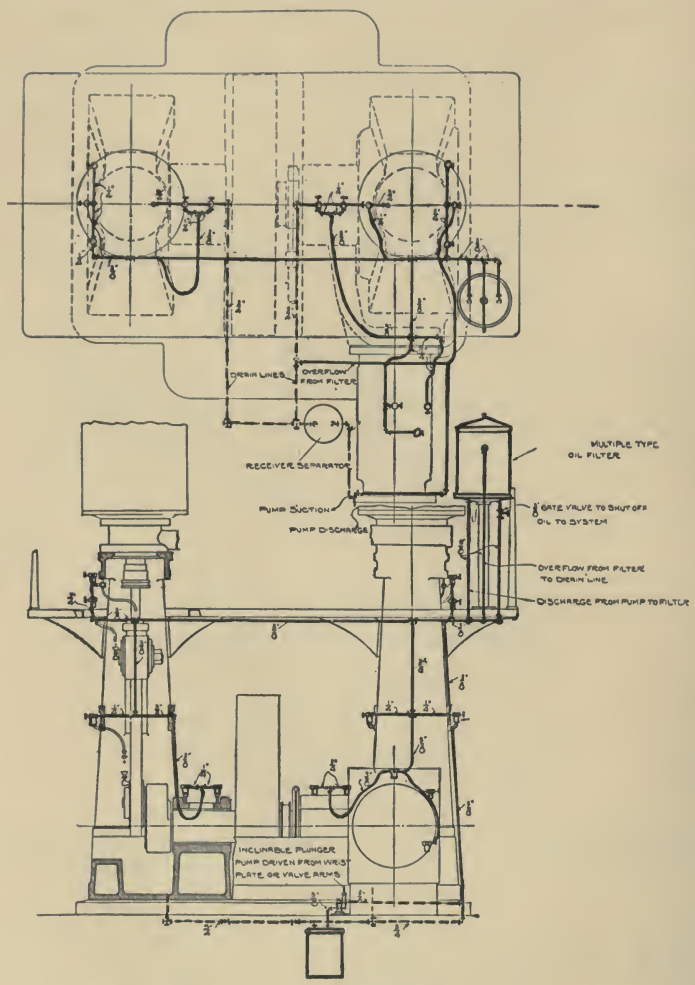


Fig. 15. The *ribbon* type of valve and a typical installation within the ammonia cylinder. [Note the thin steel strip, and the cage used to prevent excessive movement]

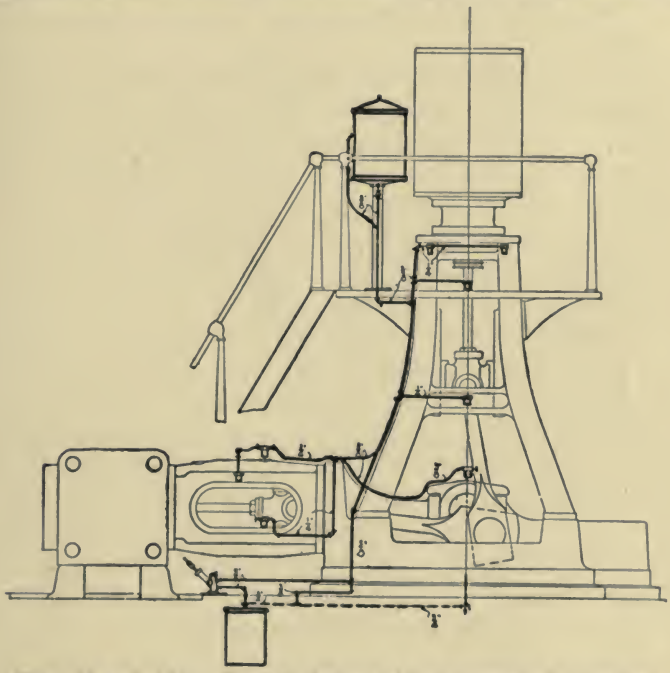
work its way into the expansion (cooling) coils, and, as the temperature is much reduced there, will most likely form a muddy paste on the sides of the pipes. This paste is composed of congealed oil, mill scale, dirt of various sorts or oxide of iron, and these accumulations will quickly decrease


DRAIN LINES
OVERFLOW FROM FILTER
RECEIVER SEPARATOR
PUMP SUCTION
PUMP DISCHARGE
MULTIPLE TYPE OIL FILTER
GATE VALVE TO SHUT OFF OIL TO SYSTEM
OVERFLOW FROM FILTER TO DRAIN LINE
DISCHARGE FROM PUMP TO FILTER
INCLINABLE PLUNGER PUMP DRIVEN FROM WRIST PLATE OR VALVE ARMS

Figs. 16 and 16A. Typical designs for Automatic lubrication applied to the open type of vertical single acting compressor

the ability of the pipe to conduct heat. With modern high speed or even moderate rotative speeds the lubricating system for the cylinder is very important and defects of lubrication, in fact, are the cause of a large part of the operating difficulties.

As already mentioned, the enclosed type of compressor uses



Figs. 16 and 16A. Typical designs for Automatic lubrication applied to the open type of vertical single acting compressor

the *splash* system of lubrication, which works with entire satisfaction on most occasions, especially if the proper kind of piston rings is used. The best conditions are secured by the selection of some modification of the gas engine piston ring, one that will positively prevent the oil from passing by the piston. However, an oil that can stand the temperature

of the suction pressure without congealing and a discharge temperature of 250 deg. F. or over without vaporizing and at the same time will not absorb ammonia, has *not* been found.* At times, especially at reduced suction pressures if the pressure drop is sudden, the oil in the crank case will be made to foam by a partial evaporation of the ammonia out of solution and the lubricant will in most likelihood pass through the suction valve.

In the case of the vertical single acting compressor the lubricant is fed into the suction line, and the stuffing box. This may be done by force feed, either by a force pump operated by hand or by means of a cam on the shaft. The horizontal machine, as a rule, has a forced feed system of cylinder lubrication—with a hand pump for occasional extra oil—and it is piped so as to enter the suction bends, or the cylinder at about the middle of the barrel. The lubrication of the horizontal main bearings, crank pin, cross head, and other parts is now usually managed by means of one of the automatic sight feed oiling systems, using an oil filter and a pump driven by means of a belt or a separate crank or eccentric from the compressor shaft. By this means the oil may be kept in good condition, and cooled by the cooling water used in the filter. The cylinder oil which is carried out of the cylinder by the compressed gas is supposed to be collected by the oil trap. It is *not* advised that reclaimed cylinder oil be used again in the cylinder although it may be used (after filtering) for other purposes. The oil trap is located near the condenser—in fact as close as may be practical—in order that the oil may be reduced in temperature as much as possible before it reaches the separator. A typical piping layout for compressor lubrication is shown in Fig. 16.

Multiple Effect Compression. For some time refrigerat-

*Cylinder oil for ammonia should not congeal at temperatures of from minus 15 deg. F. to zero deg. F., depending on the suction pressure carried, and should not have a flash point below 250 deg. F. Only straight, well refined, filtered, petroleum oil should be used.

ing engineers have advocated a segregation of the compressor work if more than one temperature was to be carried,—for instance, in cold storage where one set of rooms was desired at 33 to 34 deg. F. and another at or near zero deg. F. In ice making where the brine is held at about 14 degrees the water for the cans should be precooled to about 33 to 34 degrees. In fact a large majority of applications of refrigeration require more than one temperature and with the usual installation it means that the entire tonnage must be carried by the back pressure which is required for the *lowest* temperature desired. As the capacity of the compressor is, roughly, directly in proportion to the absolute suction pressure it is readily seen that the capacity of the plant suffers if one machine only is in operation and that at the lower pressure. One common way (if two machines are not available—one for each pressure) is to use the separate cylinder of the horizontal duplex, or of the twin vertical single acting machines for the two pressures.

Another method, one agitated for many years, has recently been adopted by the Carbondale Machine Company of using two pressures in the same cylinder. The compressor using this cycle will draw in the gas from the lowest temperature coils during the ordinary suction stroke, and at the end of the stroke the piston uncovers a set of ports in the cylinder connected with the higher pressure expansion coils. The higher pressure gas will flow into the cylinder and will increase the pressure above that of the suction stroke. A greater weight will therefore enter the cylinder per stroke which will cause a *greater* weight of ammonia to be boiled in the refrigerating coils.

To illustrate the principle (Fig. 17), suppose gas were drawn into the cylinder along the suction line *bc* and at the end of the suction stroke (at the point *c*) the high pressure suction valves were uncovered, and gas at a pressure corresponding to the point *e* were allowed to enter. The action

of this higher pressure would be such as to raise the pressure in the entire cylinder to that shown by the point *e* along the line *ce* (this is assuming that sufficient time is allowed). Instead of compression along the line *cd* the compression line will be along the line *ef*. If *c* and *e* are 15 lbs. gage and 30 lbs. gage respectively the weight of ammonia compressed will be 47 per cent greater in the latter case, and the increase in power required per stroke will be only 33 per cent (not allowing for losses).

Although the use of two or more suction pressures seems revolutionary, yet the principle of the action of the valves is

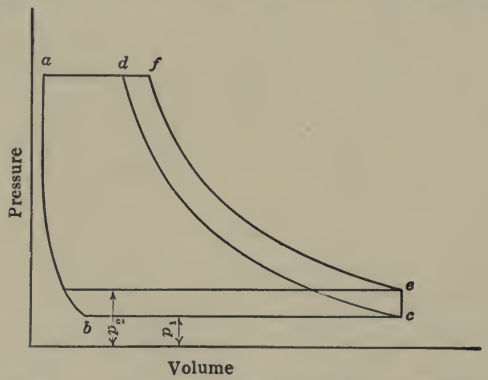
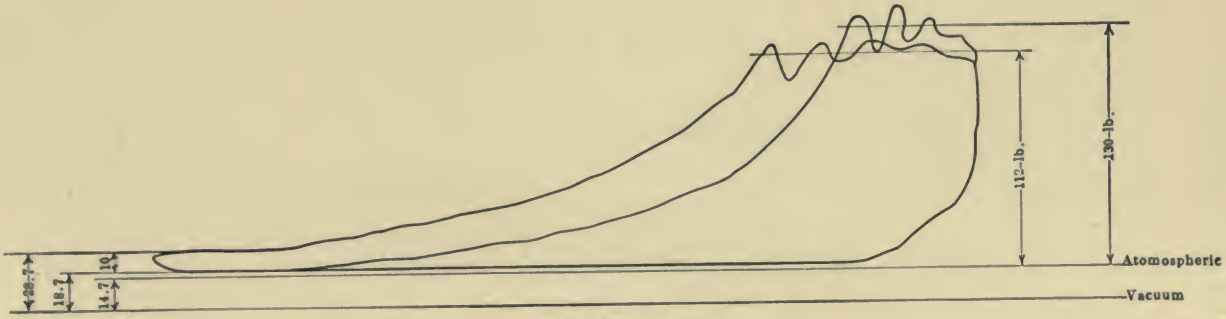


Fig. 17. Theoretical Diagram for compression with two suction pressures

not so, being somewhat of a reversal of a number of types of heat engines. The charge in a two cycle gas engine is brought into the cylinder frequently by the uncovering of cylinder ports by the piston, an operation which allows the fresh charge, at an initial compression of from 3 to 5 lbs., to pass into the cylinder. Use of cylinder ports to discharge the exhaust vapor is made in the case of the uniflow steam engine. It may be mentioned, however, that at moderate speeds the time interval is not very great for the proper flow of the gas under the higher pressure. Whereas the lower suction gas



Spring 100 Lbs. per Inch
Speed 192 R.P.M.

Fig. 18. Actual card showing multiple effect compression. [The figures are pressures scaled from the original card]

has the entire suction stroke to fill the cylinder, this other suction gas must enter during 1/6 or even 1/10 of this time interval. If a moderate speed (for the enclosed type of compressor) of 150 r.p.m. is carried, the time allowed for the process *ce* would be only 1/30 of a second at best. Such an action is possible with large valve opening, and with a sacrifice of a certain amount of pressure in pressure drop in passing into the cylinder. This seeming to be the case, it would appear that the advantage in multiple effect compression is limited to low speeds. An actual card is shown in Fig. 18.

The Carbon Dioxide Compressor. It so happens, that in the United States, contrary to European practice, the ammonia refrigerating machine is almost the only kind used—whereas the carbon dioxide compressor is very extensively used in Great Britain and on the Continent. The reason for this is hard to assign, except possibly because the carbon dioxide machine has not been pushed by any of the large refrigerating manufacturers in this country, and its advantages have not been sufficiently advertised.

As a matter of fact the carbon dioxide system has certain important advantages, and there is no question but that the future will see a great advance in its use. During the World War considerable progress was made in its manufacture here, especially for marine installations, and during that time nearly all of the large manufacturers were turning out carbon dioxide compressors.

The chief difficulty in regard to the use of carbon dioxide, perhaps, is a mental one. There is an idea that it may be used only when *cold* condensing water is available. Besides this the engineer does not like the heavy, almost hydraulic pressures encountered. Condenser pressures of 800 to 1,000 lbs. per square inch are usual, with suction pressures ranging from 200 to 400 lbs. These heavy pressures require special material in the compressor, and fittings and pipe must be

capable of taking the maximum pressures with safety and without appreciable leakage.

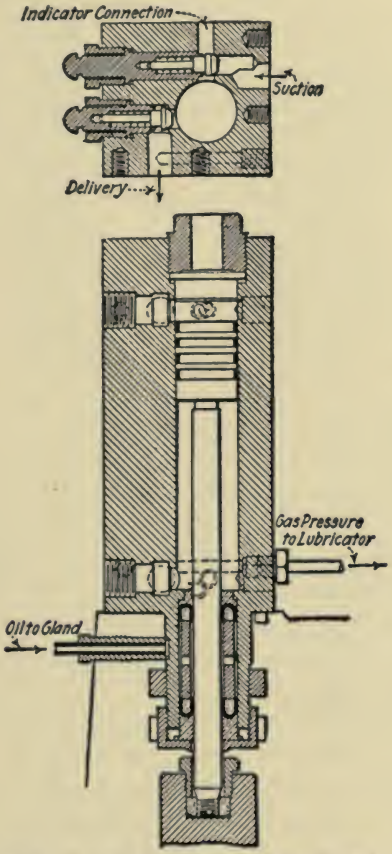


Fig. 19. The Carbon dioxide double acting compressor. [Note heavy cylinder construction and small diameter compared to the stroke]

The *advantages* in the use of the carbon dioxide system are two in number. Carbon dioxide is not a *noxious* gas in the sense that ammonia is. Quite a large proportion of the

gas can be present in a room without discomfort, as it is a neutral very much in the same sense that nitrogen is—by diluting the oxygen of the air in the same proportion. Of course the presence of too large an amount of carbon dioxide will mean the absence of oxygen, and suffocation would be experienced after a few minutes. However, the breaking of a cylinder head, or of a pipe in the line would not be as dangerous, or as disturbing to the people outside the engine room, as is the case with ammonia. Although ammonia in most cases is not explosive (certain particular percentages of mixtures of ammonia and air will burn with *explosive violence*) yet there have been certain cases of explosion and fire. Carbon dioxide is not a *combustible,* and because of its heavy working pressures it has a very small specific volume, a fact that results in a large refrigerating effect per cubic foot of piston displacement. Referring to Table 7 in Chapter VI, it is seen for the particular case chosen—of zero degrees boiling temperature of the refrigerant—that ammonia requires 5.07, sulphur dioxide 13.7 and ethyl chloride 32.5 times the piston displacement that carbon dioxide does. The compressor using carbon dioxide, therefore, will be the smallest of all known refrigerating machines with the additional advantage that copper, or brass, may be used in the condenser—a very important factor for marine work which has salt water for condenser purposes, and also because of the much greater heat transfer in the process. Fig. 19 and 20 show typical compressors of this sort.

Compressors Using Other Refrigerants. As mentioned, and as shown in Table 7, *sulphur dioxide* has a much larger volume per unit of refrigeration than has either ammonia or carbon dioxide. Except for the rather low pressures experienced there does not appear to be any particular advantage in its use, and for this matter *ethyl chloride* has but one-half the pressure of sulphur dioxide. Although a reciprocating compressor would be bulky with either of these refrigerants,

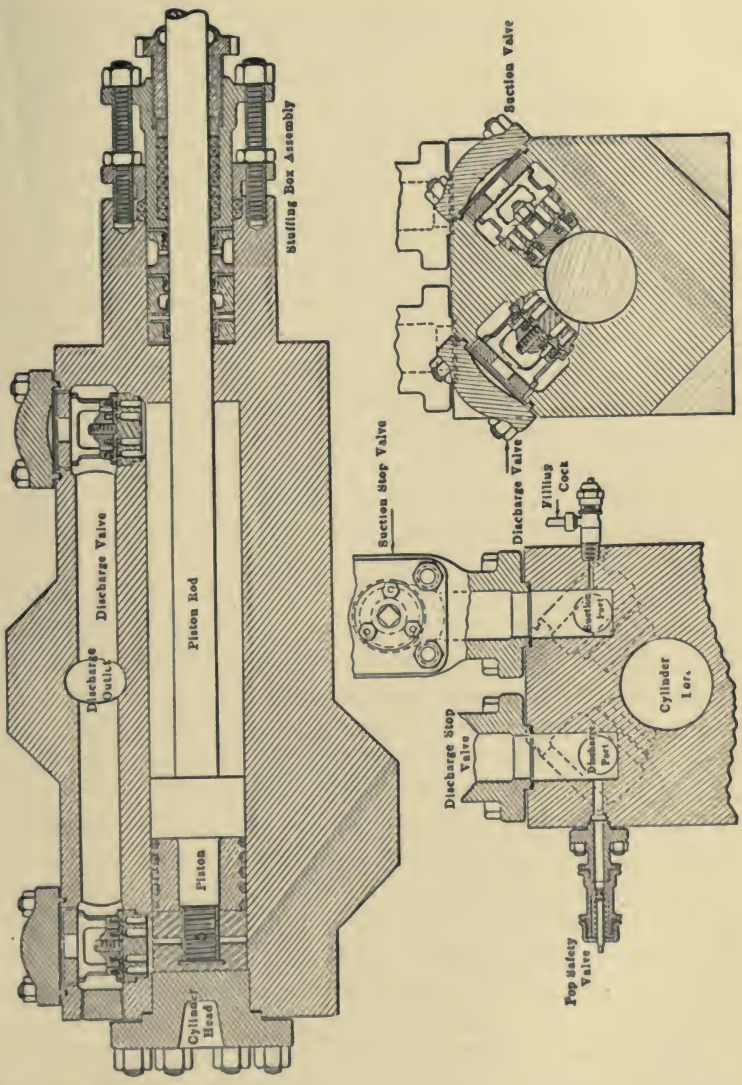


Fig. 20. Wittenmeier carbon dioxide compressor showing plate valves. [Note relatively long stroke compared with the diameter]

yet it is possible to use a rotary compressor successfully as is done in the small household "Isko" machine using sulphur dioxide, and the "Clothel" compressor used for marine refrigeration quite extensively and operating on ethyl chloride. The large volume of the refrigerant is managed satisfactorily by having a high speed rotor,—somewhat similar to the manner in which low pressure steam turbines are operated. With these two refrigerants, each being of rather heavy density, and having very nominal condenser pressures, the use of direct connection to electric motors or steam turbines should work out satisfactorily. Another successful design using sulphur dioxide is the *Audriffren-Singrun machine.* This is self-enclosed and always uses the indirect system of brine refrigeration. The design is one where the compressor, condenser and the expansion surfaces are all enclosed, sealed at the factory and non-accessible for repair. Its greatest advantage is in the fact that its operation is almost absolutely foolproof.

On analyzing the foregoing it may be said that there is a bright future especially for the carbon dioxide machine. It would seem that the only real difficulty is in getting its use appreciated. The power required for operation is practically the same for the four refrigerants given in the table, and therefore there is no relative advantage for any one above the other as to the cost of power, for nominal suction pressures.

The High Speed Compressor. As all engineers know, the four valve steam engine, and particularly the Corliss type, is the most efficient of the standard reciprocating steam engines, and yet there are many medium and high speed engines that have a ready sale for all kinds of work and locations. The reasons for this substitution lie in the fact of smaller size and smaller first cost of the machine itself. It is not expected that so high a prime mover efficiency will be obtained, but these considerations, as regards lesser floor space required, lesser engine room volume, possible use for the ex-

haust steam for heating purposes, and lower initial cost of the installation, overbalance the disadvantages.

In the refrigerating field the conditions are not exactly the same. A score of years ago rotative speeds of 30 to 40 r.p.m. were usual, then came speeds of 60 to 70 and finally 100 r.p.m. for the larger and 140 to 160 for the smaller machines (usually the twin vertical type). Of late years attempts have been made to increase the speeds still more, and encouraging successes have been obtained—the main difficulties encountered being mechanical ones due to poor lubrication or faulty machine design. Defects which cause wrecking of the machine and loss of life are infrequent. Designed properly so as to maintain proper lubrication and so as to be amply strong, the future of the high speed ammonia compressor seems bright, especially as the high speeds make it possible to connect directly with synchronous motors in sizes of fifty tons and over. As will be seen later, there is no appreciable loss in economy at high speeds provided a good form of valve is used and large area of valve opening is provided.

It has been shown that the refrigerating duty is dependent on the weight of ammonia pumped into the condenser and evaporated in the refrigerating coils. For any particular set of conditions the weight of ammonia condensed varies with the piston displacement; but the volumetric efficiency varies with different conditions of head and suction pressures, the quality of suction gas and the amount of clearance. For example the *real volumetric efficiency* (Table 1) may be anything from 80 per cent to 50 per cent or less. If the apparent volumetric efficiency (the ratio of the effective stroke to the actual piston stroke) remains the same, it means that the capacity of the machine (the tonnage) may be increased by speeding up the compressor.

Ammonia compression, as already mentioned, is similar to air compression, and the same design, with suitable changes, can be used for either kind of work. However, mechanically

operated valves for refrigeration have never been a success, on account of lack of constant conditions as regards both suction and discharge pressure, and also because of the additional difficulty due to having the necessary valve stems pass through the cylinder walls. The valves are therefore gas operated and automatic of one type of design or another.

TABLE 1. HORSEPOWER PER TON OF REFRIGERATION [AMMONIA]

Condenser Pressure Lb. Gage	Suction Pressure Lb. Gage 5	10	15	20	25	30
145	2.02	1.69	1.40	1.26	1.11	0.99
155	2.11	1.76	1.48	1.38	1.19	1.05
165	2.21	1.85	1.57	1.41	1.25	1.13
175	2.34	1.94	1.65	1.48	1.34	1.18
185	2.48	2.03	1.75	1.57	1.40	1.27
195	2.55	2.14	1.84	1.67	1.48	1.33
205	2.69	2.24	1.91	1.75	1.55	1.40

VOLUMETRIC EFFICIENCY OF AMMONIA COMPRESSORS

Condenser Pressure, Lb. Gage		Suction Pressure Lb. Per Sq. In. Gage 0	10	20	30	40
120	A.	0.77	0.83	0.87	0.89	0.91
	B.	0.60	0.70	0.77	0.81	0.84
	C.	0.52	0.65	0.72	0.77	0.80
160	A.	0.74	0.80	0.83	0.86	0.88
	B.	0.54	0.65	0.72	0.76	0.80
	C.	0.44	0.58	0.66	0.72	0.75
200	A.	0.71	0.77	0.81	0.84	0.86
	B.	0.49	0.61	0.68	0.72	0.75
	C.	0.37	0.52	0.62	0.67	0.71

A, No clearance. B, 4 per cent clearance. C, 6 per cent clearance.

As regards the type of design of valve for high speed the valves must be of special form. The reason for this is easily seen on considering the problem. It will be remembered that the suction and the discharge valves are opened and closed by the action of the difference of pressure on the two sides of the valve. In the case of the suction valve the pressure must be less on the inside than the outside of the valve, and the discharge valve requires a greater pressure in the inside than on the outside of the cylinder in order to open it. This

difference of pressure increases with the weight of the valve and the speed with which it is opened (the effect of inertia). Again as the piston speed increases, the size of the valve openings must increase so as to allow the gas to pass in and out with the least possible effort, and thereby keep the so-called velocity head down to a minimum.

The effect of inertia can be seen by common examples: A boat in the water, or a train of cars, requires some time at full power to attain even moderate speed. A paper may be jerked from under a book, which if moved slowly could have carried the book along. The skater or the car starting from rest requires full power to attain speed quickly. These examples are all subject to the law that the force required to change the velocity of a body varies as the product of the *weight* of the body and the *change of velocity.*

So it is in the case of the valves of ammonia or air compressors. As the speed increases, the force necessary to open or close the valves quickly becomes very great. The sudden application of the force necessary for the purpose tends to weaken the material also. For these reasons the demand is for a type of valve that is light in weight; a demand that has been met by means of the ribbon, feather or plate valves.

As has been already mentioned, the valves used for ammonia compressors are those which have found success in their application to air compression, both in moderate pressures and in the nominal pressures required for blast furnace compression. One of the most widely known of these is the *plate valve.*

The plate valve, as its name suggests, usually is made of a thin sheet of steel, is hardened, and is ground down to a correct flat surface. In the older types—the Borsig and the Gutermuth valves—and the more modern valve—the Mesta—flexibility is obtained by perforating the plate (after the principle of a spiral) so that the movement of the plate is easy of accomplishment.

The main objection to such a design is the fact that the valve is fastened to the seat by means of a machine screw attached to a central part of the spring. This means that the valve lifts off its seat by a bending action similar to the deflection of a cantilever. The operating result is trouble due to occasional breaking on account of repeated stress along the position of maximum bending. Although trouble of this sort is reduced by heat treatment and by selected material used in the plate valve, and is only of slight difficulty in the case

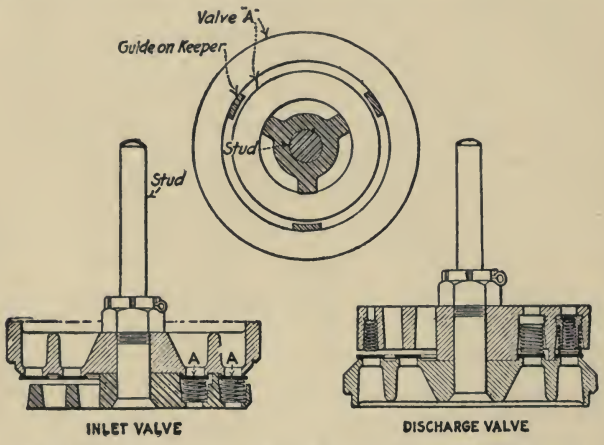


Fig. 21. The modern type of plate valve, with closure by means of a spring. [Note the thin, hardened steel ring ground to a true surface]

of air compression, yet for ammonia this type of valve has not been found entirely satisfactory. One of the latest of the plate valves is a distinct variation from previous plate valves. It is more like a disc and is flexibly held on its seat by means of a spring. In consequence there is no call for deflection of the material, and no chance of fracture on that account. A number of manufacturers are using designs similar to Fig. 21 for both air and ammonia compression. The valve lifts off its seat more like a disc valve in the water end of a pump.

The ribbon valve has, without question, proved its value in refrigeration work. At first applied to refrigeration only in several special designs it has now been adopted by one of the refrigerating companies for all its horizontal machines.*

There seems to be a difference of opinion in regard to the relative advantages of the various types of valves. The main objection to the use of the plate valve seems to be in the alleged large resulting *unbalanced pressure.* Referring to Fig. 21 it will be seen that the plate valve has to overlap the ports enough to secure a firm seat. This amount may be 1/16 of an inch all around the port. The result is that some engineers claim that an unbalanced pressure is present, due to the much greater area of valve on the top as compared with the area subject to vapor pressure on the underside of the valve. This difference of area would appear to be from 25 to 35 per cent of the port opening. If such a condition prevailed the compression line would have to show an excessive pressure rise in order to open the valve. Although this contention is true in some cases it certainly is not generally so. In the horizontal compressors using the plate or ribbon valve the suction and discharge valves are identical, and frequently of the same size, and in some cases the two are interchangeable by the simple expedient of reversing them.

Effect of Clearance. For some time we have heard of the dire effects of clearance, and how it should be reduced to a minimum. It is true that clearance will reduce the capacity of the compressor, but it is not true that four or five per cent of clearance will increase the horse power per ton of refrigeration very appreciably. The superheated gas remaining in the clearance space at the end of the stroke expands at the beginning of the suction stroke and does *work* on the piston. This work is nearly equal to the work done *on* the gas during

*Recently it has been announced that two companies identified with the manufacture of absorption machines for some time are to build ammonia compressors using plate or ribbon valves.

compression—theoretically it is equal. On account of this re-expansion, however, the effective stroke of the compressor is reduced due to the reduced volumetric efficiency. Additional capacity is secured by increasing the piston displacement—increasing the cylinder diameter or by increasing the rotative speed—by an amount which is not very great provided that the condenser pressure is normal. There is also the factor of the compressor friction, which becomes greater in proportion with decreased volumetric efficiency. Incidentally the plate form of valve is one which usually requires considerable clearance volume, because of the cages used to secure it accurately in position and to enable it to function properly, and therefore four to seven per cent of clearance will be found in all such compressors.

Low Temperature Refrigeration. Certain applications of refrigeration require low temperatures, the production of which adds to the difficulties and operating problems of the engineer. These low temperatures are found in cold storage work, possibly in ice cream hardening, in oil refining and in special chemical industries of various sorts. In such applications of refrigeration, temperatures of zero degrees Fahrenheit or lower are maintained, thus requiring minus 15 deg. F. or lower boiling temperatures of the ammonia.

At *low boiling temperatures* the ammonia compressor of the standard design is not efficient except at low head pressures. At normal condenser pressures the standard machine has the following capacities per ton, 20 lbs. suction and 175 lbs. head pressure being taken as unity:

Suction Pressure	Capacity
40 lb. gage	1.611 tons
30 lb. gage	1.306 tons
20 lb. gage	1.000 tons
10 lb. gage	0.699 tons
0 lb. gage	0.410 tons
—5 lb. gage	0.263 tons

From this table it is seen that at low suction pressures the standard compressor fails as a medium for securing refriger-

ating duty, and the engineer sees clearly why it is necessary to designate the working conditions when stating the capacity of the compressor, *e.g.*, 5 deg. boiling temperature and 86 deg. condenser temperature. In the figures given a 100 ton machine at 20 lbs. suction and 175 lbs. condenser pressure would have only 26.3 tons capacity at a pressure of —5 lbs. gage.

The reasons for *lowered capacity* with decreased suction pressure have already been brought out but it is worth while making mention of them again at this time. These reasons are the low density of the ammonia at low pressures and the small volumetric efficiencies obtainable. In addition there is trouble in the condenser on account of overheating the ammonia.

Some operating engineers have no clear idea of the so-called *density* and its effect on the tonnage of the machine. Yet in the use of steam they see clearly the reason for the relative increase in size of the exhaust steam pipe as compared with the high pressure steam pipe supplying the steam —especially when a low vacuum is carried as in steam turbine practice. In the steam engine or turbine the same weight of steam is exhausted as enters from the steam header, but the pressure is much *lower*. We say under these conditions that the *volume* of a unit weight of steam (one pound) is much greater with the lowered pressure. It is like pumping air into a bicycle or automobile tire—the air under pressure has the same *weight* as before being compressed, but it has only 1/5 or 1/6 of the volume. From these comparisons it will be seen that the density of ammonia is reduced as the pressure is lessened as will be even more apparent from the table of approximate values:

Pressure, lb. gage.	Density of ammonia, cubic feet per lb.
20	8.0
15	9.3
10	11.0
5	13.7
0	18.0
—5	26.0

The ammonia compressor, which has a fixed piston displacement per stroke, will pump the same number of cubic feet of ammonia per minute, but will have fewer and fewer *pounds of ammonia discharged* out of the compressor as the pressure is lowered. And the tonnage is always proportional to the amount of ammonia *in pounds* which is pumped into the condenser per unit of time, usually taken in minutes.

The *volumetric efficiency* has to be considered because of the re-expansion of the gas in the clearance space at the beginning of the suction stroke, a matter which is a very important consideration when there are great differences in the suction and discharge pressures. This re-expansion prevents the inlet valve from opening until the gas in the clearance space has been reduced at least to the suction pressure. Under conditions of operation prevailing in low temperature operation this re-expansion factor looms in large proportions and materially affects the effective stroke. Clearance re-expansion, as already considered, is not a direct cause of increasing the power required to compress and pump a given amount of gas, but it cuts down the capacity of the machines as the cylinder is supplied with much less than the full cylinder volume of gas per stroke.

The *third objectional feature* of the single stage compression as applied to ammonia is the high temperature of discharge of the gas. In air compression the rule is usually to pump from atmosphere to about 70 lbs. in the first stage, and from this pressure to the required final pressure when moderate pressures are required. An intercooler is placed between the stages, and the gas is cooled to within a few degrees of the available cooling water before passing on to the second stage. The reason for so doing is to increase the volumetric efficiency, decrease the work of compression and prevent *excessive discharge temperatures.*

Excessive discharge temperatures are bad for ammonia. Considerable difference of opinion has been expressed on the

subject, but it is clear from the practical experience of refrigerating engineers that high discharge temperatures, as well as high discharge pressures, tend to disintegrate the ammonia into hydrogen and nitrogen as well as to cause excessive vaporizing of the lubricating oil. These disintegrated gases are permanent, and they collect in the condenser and—unless purged—gradually increase the condenser pressure in a manner similar to the way that air increases the pressure in a steam condenser when the air pump is stopped. The formation of inert gases in the condenser has resulted in excessive purging and heavy losses of ammonia from passing out of the condenser during the purging process.

Parallel with the development of the ammonia compressor came the *absorption machine* (which uses heat energy directly instead of mechanical energy, to return the ammonia gas to the condenser again). The absorption machine uses any form of heat energy, but at present exhaust steam is most used. By the use of the absorption machine—which by a special device does not have to handle the ammonia gas directly as an exhaust gas—low temperatures may be readily and economically obtained at the expense of about 30 lbs. of steam for 10 degrees brine and 34 lbs. of steam for minus 10 deg. F. brine, calculated for 60 degrees cooling water. At first it would seem that the objection to low temperatures does not hold with the absorption machine, and in fact for some time it was considered that the low temperature industry was the especial field of the absorption machine, and that the compressor could in no way compete with it. This contention is no longer true, although the absorption may still hold its own under particular conditions.

The absorption machine has never been very popular even at the best time of its career. It is large, costly, awkward and more difficult to operate successfully than the compressor. Steam must be available. When operated by competent engineers, however, it is satisfactory, and can achieve low tem-

peratures with ease. Until the perfection of the stage ammonia compressor it was used for all low temperature work.*

Stage ammonia compression for low temperature refrigeration has a number of important features. It is compound compression, with an *inter-cooler* between the stages. It is possible to arrange the suction of the high pressure so that it will take care of the gas from the freezer rooms—as is done in multiple effect compression—but this scheme has its limitations. It is more advantageous to design the cycle in a manner similar to the compound steam engine, and to try to equalize the load due to the compression between the two cylinders. Thus, instead of having a receiver pressure of 25 or 30 lbs. gage, a pressure of some 45 to 55 lbs. would have to be used, depending on the suction and head pressures carried.

As we all know, compound compression requires an intercooler as well as an aftercooler (the condenser in refrigeration). In air or ordinary gas compression water is used in the intercooler, but this is not possible with ammonia because of the low temperature of saturation of 30 to 38 deg. F. at 45 to 55 lbs. gage respectively. It is evident that water cannot be used for such cooling, unless the intercooler used water for the first part of the cooling process and finished by means of some other cooling medium similar to the practice in the Baudalot milk cooler. The only convenient and natural medium for such low temperature intercooling is ammonia. This is done by means of an *accumulator* arrangement.

The functions of the accumulator are twofold. It will be remembered that under usual circumstances there is some 10 to 15 per cent of ammonia vaporized at the expansion valve in order to cool the remaining liquid ammonia from the temperature of the liquid receiver to that of the tem-

*Chapter IV is devoted in part to a detailed study of the absorption machine.

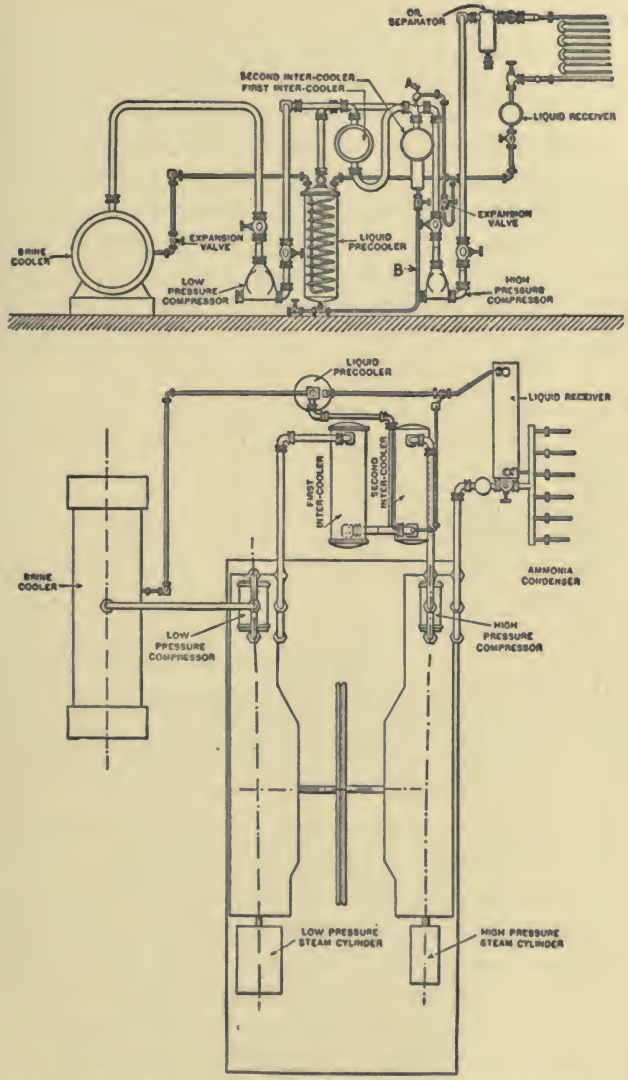


Fig. 22. The Vilter Manf. Co. [D. I. Davis] method of securing stage compression

perature of the boiling liquid in the expansion coils. The vapor formed in the process cannot be used for refrigeration, but clogs up the expansion system unless removed. The gas formed increases in amount in cases of extreme pressure range, as in the case where low temperature refrigeration is required, and so the accumulator principle is used to remove the gas formed in pre-cooling and to remove the superheat from the discharge gas from the low pressure cylinder.

Fig. 22 shows the general arrangement of the cylinders

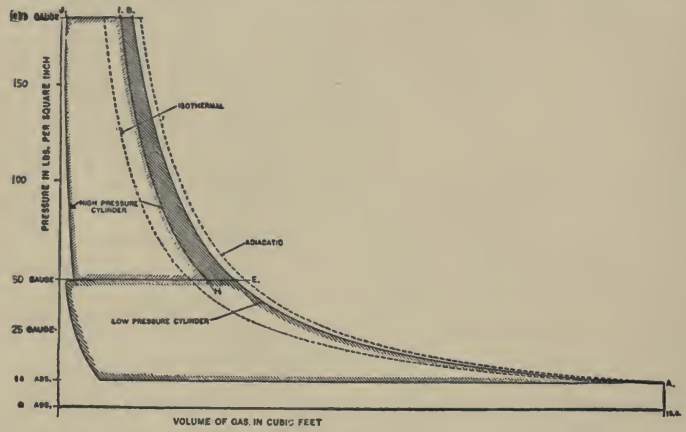


Fig. 23. Stage ammonia compression. [The theoretical card]

and intercoolers. Referring to the figure, it will be seen that the gas discharged from the low pressure cylinder is made to go through the water-cooled first intercooler and then through the ammonia-cooled intercooler. The discharge gas is thereby cooled from, say, 100 to 150 deg. F. to about 35 deg. F. and some 5 per cent (perhaps) of liquid ammonia is vaporized in the process. Liquid ammonia not vaporized after passing through the second intercooler through the valve A, flows by gravity down pipe B into the liquid precooler which is open at the top to the intercooler

receivers, and the gasefied ammonia passes out and enters the high pressure cylinder of the compressor. The liquid from the liquid receivers, having passed through a long coil in contact with boiling ammonia, will be precooled to about 30 to 35 deg. F. or any other temperature desired.

But what is the advantage of all this? It seems that a lot of extra piping has been designed all to no purpose, as all these things will occur anyway. Yet this is not true, as can be seen by a careful consideration of the conditions which prevail. It is true that gas is formed by cooling the liquid

Fig. 24. Stage ammonia compression, taken from actual cards

in the liquid precooler, but this gas is compressed only in the second stage, thus decreasing the work necessary to be done in the low pressure cylinder. Some liquid is evaporated in the second intercooler, but the volume vaporized is only one quarter of the decrease in volume of the gas passing into and out of the intercooler. Finally the temperature of discharge from the second stage compressor is only about 175 deg. F. (point I) instead of 350 deg. F. (point B) in Fig. 23. The clearance re-expansion loss is reduced, as seen in Fig. 24, which is taken from an actual indicator diagram.

It is seen from the diagram that the two compressors are driven from the same shaft, which may be engine, synchronous electric motor or belt driven. If desired, separate drives may be made, as is done in the Central Manufacturing Company cold storage warehouse, of Chicago, where large units are employed. In this plant the low pressure cylinder is called a "booster" and liquid cooling is used in a somewhat similar manner to that already described, which is known as the D. I. Davis method. In moderate sized plants, however, the advantage of having the two cylinders on one shaft lies in the reduced torque and decreased flywheel and balancing troubles.

In summarizing this discussion of compressors especial attention might well be called to the three main types of compressor design, the vertical single acting (Fig. 16), the horizontal double acting (Fig. 11), and the enclosed type of compressor (Figs. 8 and 10), since these are the chief fundamental designs, and represent very nearly present day construction in the United States. But the compressor is only a small, though an important, part of the refrigerating cycle. The other parts are the condenser and its connections and the expansion (low pressure) side, with its fittings and auxiliary apparatus, and these now will be considered in Chapter III.

CHAPTER III

Condensers, Ammonia Fittings and Accessories

Sometimes in taking up a new subject, or an old one from a new point of view, it is worth while to stop occasionally and consider in retrospect the ground already covered. This idea of "mulling" over problems until the solution has become clear is worth while for every one. For example, no engineer who has mulled over the principles involved in making steam in a boiler, and the vast heat developed in the combustion of the fuel as compared with the relatively small amount of steam generated, could ever believe the rather common explanation that ammonia "flashes into a gas" on passing the expansion valve. Of course some gas is generated, but only about 10 or 15 per cent, in a manner similar to that experienced on the occasion of opening the lower cock of the water column on the steam boiler.

And therefore it is wise to look backwards and re-examine the refrigerating cycle, and the function of the different parts of the process. Consider, if you will, that the refrigerating cycle is the reverse of the steam engine cycle; anti-clockwise rotation for example. Start with the compressor which takes a vapor that may be wet, dry and saturated, or superheated, and pumps it up to a new level (a new *temperature* level) where the new condition is always that of a superheated gas under much heavier pressure. The temperature of the discharge from the compressor varies with other conditions but is always greater than 100 deg. F. As work is

done on the gas the process is opposite to the action of the steam engine. Then the compressed gas is cooled in a condenser, to remove first the superheat, then the latent heat of condensation and finally *some* of the heat of the liquid. This action is exactly the opposite from what takes place in the steam boiler with a superheater. Finally the expansion or refrigeration coils (in which heat is absorbed from the surrounding mediums) have an action just contrary to that of the steam condenser. These *three* separate processes are necessary in order to complete the cycle, and to enable the same refrigerant to be used continuously.

Although the term "superheat" has been used several times, and is a common one in steam engineering, it is yet worth additional comment. The ordinary steam boiler gives saturated steam. Superheated steam is obtained by having a portion of the heating surface so located as to take up some of its heat *out* of contact with water in the liquid state. The ordinary water tube boilers—the Babcock and Wilcox, the Stirling, the Heine, etc.,—and the fire tube boilers—like the horizontal return tubular boilers, the Scotch marine, and the locomotive types—all give saturated steam which is usually a little bit wet. The vertical boilers of the donkey, fire engine, and the Manning type give superheated steam without (of course) the use of the superheater, because the water level in these boilers is below the level of the upper tube sheet. Likewise the refrigerating coils of a refrigerating plant will give dry saturated vapor in the suction return pipe *provided* no heat is abstracted by the ammonia (or other refrigerant) while *not* in contact with its liquid.

As an example of what has just been said suppose water is pumped into a boiler at 110 deg. F. and is evaporated into steam at 150 lbs. gage pressure. Then the water will be heated with a rise of temperature until a temperature *corresponding* to that of saturated steam at 150 lbs. gage, or 366

deg. F., is obtained and then (and only then) evaporation will begin. As the pressure is maintained constant, boiling will continue at a constant temperature of 366 deg. F. Steam so generated is given the name of saturated steam, but it is not *dry* and saturated unless some form of dry pipe is used, or the boiling action is very slow. If a superheater is used, or one of the types of vertical boilers just mentioned is employed, then the steam becomes dry and *superheated* an amount depending on the conditions prevailing. This may be enough to make the final temperature of the steam 400 or 500 deg. F.

In the case of a refrigerating plant, if the ammonia is allowed to boil at 15.72 lbs. gage (assuming a standard barometer pressure) boiling will take place at a constant temperature of zero degrees, and the ammonia gas will have that temperature, provided heat is not removed by the ammonia at any time while *not* in contact with liquid ammonia. If the refrigerating coils are similar in arrangement to that of a vertical boiler so that some of the surfaces convey *dry* ammonia then the temperature of the gas will rise (it becomes superheated) and the return gas to the compressor will be superheated and the final temperature will be 10, 15 or more degrees Fahrenheit, depending on the temperature of the substance being refrigerated and other conditions of operation. Likewise, if a long suction return pipe takes the gas through rooms held at ordinary temperatures or if the pipe is poorly insulated, then the amount of superheat will be excessive. The gas entering the compressor, then, is superheated except in the case of the older types of machines operating on the so-called wet compression.

During the suction stroke the ammonia gas is superheated an additional amount due to the heating effect of the cylinder walls and ports, which have been heated by the hot discharge gases and during compression this superheat becomes very much greater still. Frequently the amount of super-

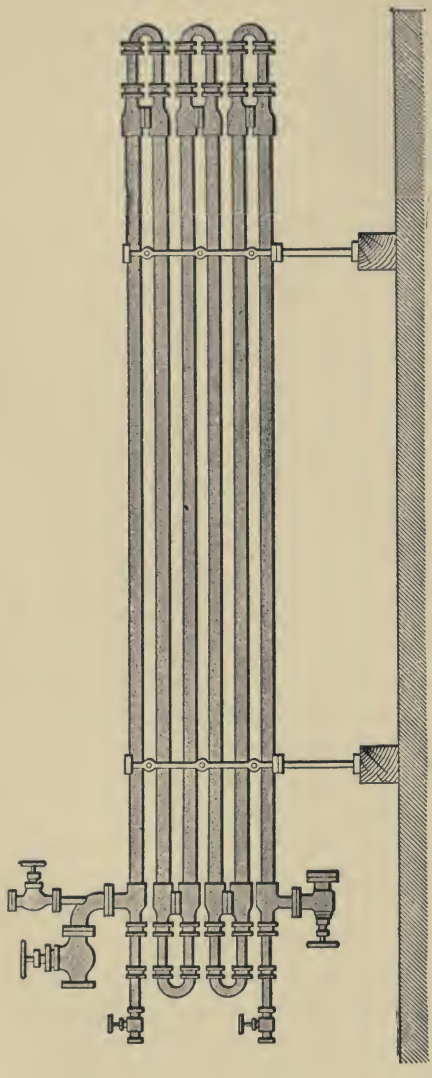


Fig. 25. The double pipe ammonia condenser, with provision for the water to pass through the inner and the ammonia through the outer pipe. Condenser designed for counter flow

heat after compression is as great as 125 to 200 deg. F., but now figured above the *saturation temperature* corresponding to the discharge pressure. The reason for this superheat in the gas during compression is because of the increase of the internal energy during compression in consequence of work having been done on the refrigerant during a process which is approximately an adiabatic.*

The Ammonia Condenser. From the preceding, and from the consideration that the condenser in the refrigerating cycle corresponds to the boiler of the steam engine cycle, just reversed in its action, it becomes a simple matter to calculate the amount of heat to be removed. The gas enters the condenser superheated and is condensed and finally cooled (as a liquid) to a point depending on the temperature of the cooling water and the form of the condenser used. Taking an example of gas entering the condenser at 165 lbs. absolute (150.3 lbs. gage) and a temperature of 250 deg. F., the saturated temperature at this pressure is 84.5 deg. F. and the total heat measured above 32 deg. F. is 661.7 B.t.u. (see the ammonia tables, Chapter VI). If cooling water is used of cold enough temperature to cool the condensed liquid to 75 degrees, then (as the total heat of liquid ammonia above 32 deg. F. is 47.8 B.t.u.) the heat removed by the cooling water per pound of ammonia under these conditions is $661.7 - 47.8 = 613.9$ B.t.u.

The function of the condenser is to remove heat at the upper temperature, that is, the discharge pressure from the compressor. As a pure liquid refrigerant is required for use in the expansion coils it is necessary to have a *surface* condenser only, one providing contact between the water and the refrigerant. However it is much more of an engineering proposition than simply providing a certain area of cooling surface, although in a certain way it is much simpler than is

*The adiabatic is a term used to denote an expansion or a compression line which does not receive heat or give up heat during the process. A compression line on a medium speed compressor is very nearly such a line.

the problem of steam surface condensers working under a vacuum.

The Effect of Air Film. In ammonia condenser practice there is always a compromise between the amount of cooling water to be used, and the cooling surface provided. If we use little water, and heat it considerably—say 15 to 20 deg. F.—then we will increase the head pressure, as the head pressure is dependent on the temperature at which liquefaction takes place. As a rule, this pressure is nearly that of the *final**

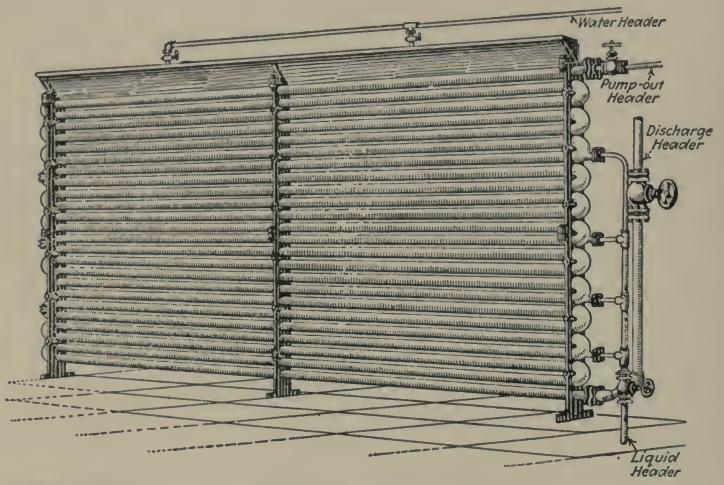


Fig. 26. The bleeder [drip] atmospheric type ammonia condenser. The action of the gas and water is counter current

temperature of the cooling water. If we use quite a little water, with a water temperature rise of 5 to 8 deg. F., then the cost of pumping water assumes larger proportions. But more important than all is the question of heat transmission. In the steam surface condenser the water usually passes through the tubes and the steam is allowed to enter the shell of the condenser in a manner such as to allow it to impinge

*The head pressure may be considered as that corresponding to the final water temperature *plus* five degrees.

perpendicular to the tubes. In this steam is a little air, which seems to stick to the tubes and form an insulation air film around the tubes. The result is a much reduced rate of heat transmission, and the effect finally is that it takes two or three square feet to do the same amount of work that would be required of one square foot without the insulating film. In the steam condenser the air is removed constantly by means of an air pump, which compresses the rarefied air to a pressure great enough to discharge it into the atmosphere, and removes during each unit of time the amount of air leakage during the same space of time. In the ammonia condenser there is always some air as well as decomposed ammonia and oil, all of these tending to the formation of a non-condensible gas film. However, here there is a difference in that—as the flow of the ammonia is along the tubes—there is a tendency to scrape off this film if there is some place to which the accumulated gas may go. For instance, in the bleeder type of condenser (Fig. 26) the gas enters at the bottom and passes upward, suitable provision being made so that the liquid ammonia may be drained or bled off as it is condensed. The idea here is, as it is in the steam condenser, that the best efficiency may be obtained by operating the condenser with as little liquid in the pipes as is possible. In the flooded condenser, the double pipe (Fig. 25), and the atmospheric types, the gas from the compressor enters at the top and works downwards so that the lower pipes are partly, or, possibly, completely filled with liquid ammonia. In Fig. 27 it is seen how this affects the action of the condenser, inasmuch as the effective area of the pipes is reduced in proportion to the area exposed to the liquid.

Effect of Oil and Scale. In addition to the effect of a gas film, or accumulated liquid on the pipe surface, there is the effect of oil and scale. This is not a question of design, except that the right kind of condenser should be chosen for the kind of condensing water at the plant. Boiler tubes re-

quire cleaning for economy as well as for safety, and ammonia condensers have a similar need. The condenser must have a ready means of being cleaned, and cleaning should be performed frequently enough to prevent the solid material from forming a hard scale.

The Temperature Difference in a Condenser. In condensers, as well as other heat transferring apparatus, there are three factors which affect the quantity of heat conducted through the walls of the pipes: the area exposed, the co-

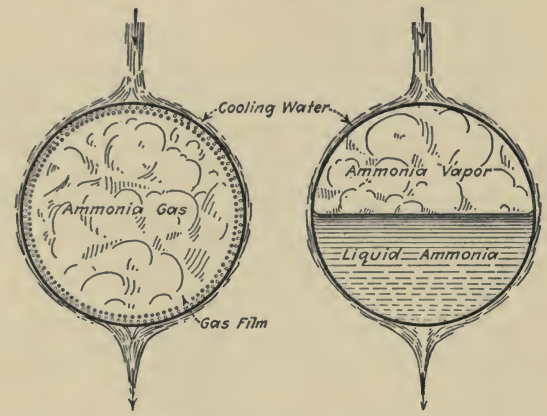


Fig. 27. The ordinary and the flooded atmospheric ammonia condenser in section, showing the probable formation of gas films and the liquid levels

efficient of heat transfer, and the temperature difference on the two sides of the pipe. The product of these three quantities will give the number of heat units transferred per hour or per 24 hours. The area is taken in square feet, and the coefficient of heat transfer is found by experiment and is affected by the kind of metal used, its thickness, the condition of the surface,—whether oily or scaly—or whether a gas film is around it. The temperature difference requires a little more explanation.

In Fig. 28 an attempt is made to show the action taking place in the condenser. Two cases are shown, one representing parallel flow of the gas and water—as in the case of the common atmospheric type of condenser where the water flows out of a trough and then falls first on one pipe and then on another,—the other is a counter flow of the gas and the water as in the bleeder type, or the double-pipe type of condenser. From the diagram it will be seen that parallel flow and counter flow do not give the same results. What is desired most particularly is to get as cold a liquid ammonia as is

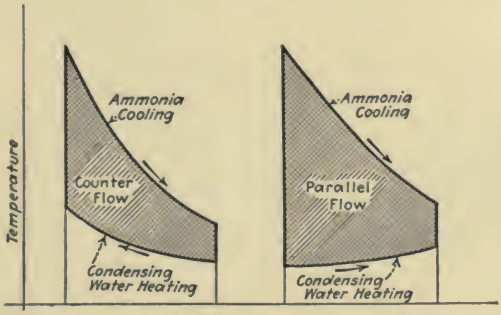


Fig. 28. The *relative action* of the condenser, as regards the change of temperature of the water and the gas. [Note that the parrallel flow does not cool the liquid refrigerant to as low a temperature as does the counter flow type]

possible, because it has been found that more refrigerating capacity is obtained thereby, and the condenser pressure is lowered. Therefore it is most desirable to bring the coldest water into contact with the coldest ammonia, and the warmest water where the compressed gas enters. To show how this works out in practice it may be said that the usual specification for the atmospheric type of condenser is 60 lin. ft. of 2-in. pipe, but only 30 lin. ft. for the bleeder type, using the same liberal factor of safety.

The Flooded Condenser. A type of condenser that was regarded with considerable favor some time ago was the flooded condenser (Fig. 29), a design which is now not so popular, on account of the difficulty in the control of the discharge pressure, and proper working of the condenser. The idea in this arrangement was that the coefficient of heat transmission in the ordinary condenser from the gas to the water is very low in value, but that if the surface could be wetted then results would be better. As a matter of fact the condenser surfaces are inherently wet because of the rapid disappearance of the superheat in the ammonia, and

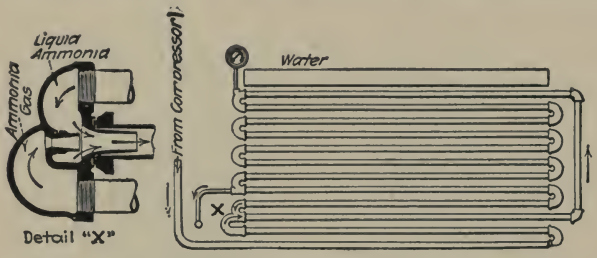


Fig. 29. The flooded condenser, showing the detail *X*, whereby the gas from the condenser picks up liquid ammonia by means of an ejector nozzle

the great difficulty with the flooded condenser is to prevent, as the name really implies, the actual choking up of the condenser tubes with liquid ammonia, so that the conduction of heat would be retarded, instead of being accelerated, if the gas velocity was not increased by choking up the pipes. The action of the condenser is to install an injector so arranged that the gas from the compressor passes through and drags some liquid ammonia from the liquid receiver and the gas and liquid mixture passes through the condenser, as in the ordinary design.

With pure water, and especially with small sizes, the double pipe condenser is the best all around design, but with large sizes and with salt water or fresh water subject to salts or

vegetable growths there is a demand for a compact condenser which may be readily cleaned. In consequence the vertical shell and tube condenser is becoming prominent, either with heads so as to have more than one pass of the water (Fig. 30) and with the tubes filled solidly with water or with open heads and by use of adjustable thimbles at the top to allow a film of water under rather high velocity to pass over the

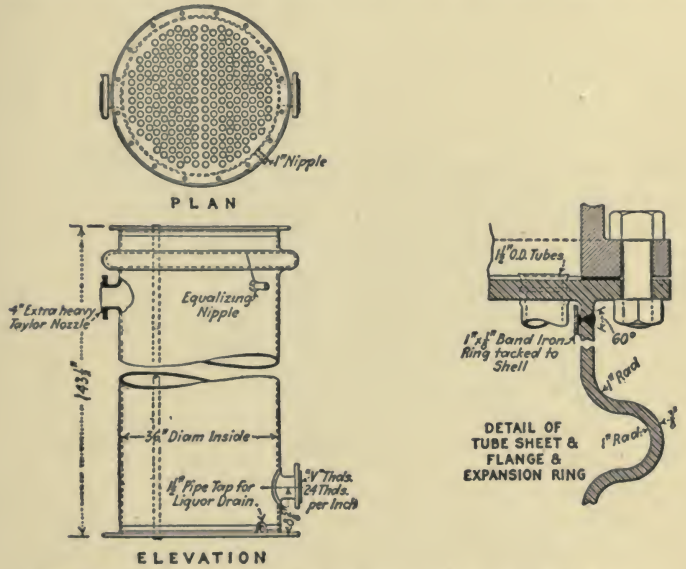


Fig. 30. A special design of ammonia condenser, somewhat on the principle of the surface steam condenser

surface. In either of these designs a very large condenser surface is possible within a small volume, the path of both water and ammonia is direct—thereby reducing friction head,—the apparatus is easily cleaned and inspected, and the condenser pressure is maintained close to that corresponding to the exit condensing water temperature.

Condenser Surfaces. As regards the capacity of an ammonia condenser, the condition is similar to that of a steam

boiler of the water tube type. It is usual to rate such a boiler at 10 sq. ft. of heating surface per boiler horsepower; yet there are cases where such boilers have developed 2, 2½, 3 or even more boiler horsepower for every 10 sq. ft. of heating surface. A boiler capacity depends on the condition of the water, of the tubes and the ability to burn fuel under the tubes of the boiler. In like manner the condenser may be

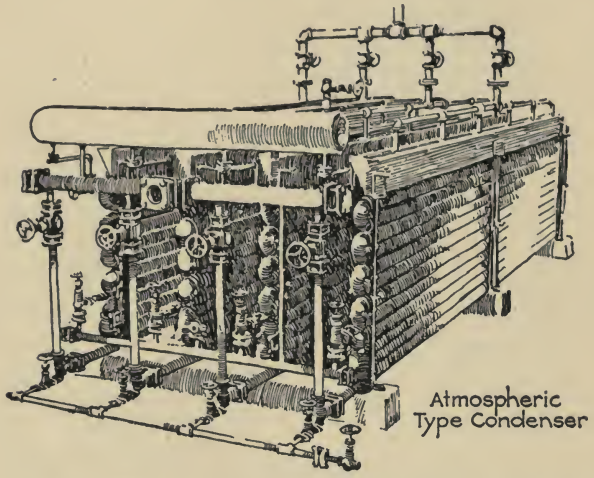


Fig. 31. The atmospheric type of condenser, showing distributing troughs for showering the water over the sides of the pipes

able to do cooling at the rate of 12, 10 or even 6 sq. ft. per ton of refrigeration, and yet we would not be justified in using such small surfaces. The efficiency of the condenser may drop off, some stands may be laid down for repair, or perhaps storage volume may be required for the liquid ammonia during times when repair work on the low pressure side is required. As a matter of fact, the usual allowance in condenser practice is 4 to 8 sq. ft. per ton of refrigeration for the flooded, 6 to 8 sq. ft. for the double pipe, 12 to 20 for

the shell and tube, 20 to 25 for the bleeder and 25 to 30 sq. ft. for the common atmospheric type of condenser.

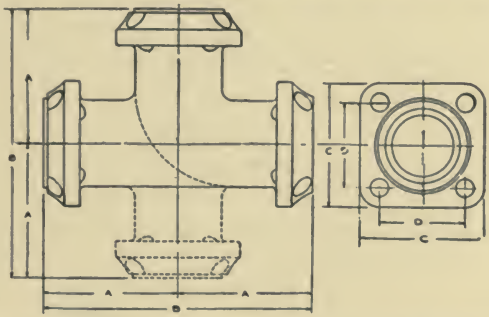
The most important function of the condenser is the determination of the condenser pressure, which determines directly the work done by the steam engine or the electric motor used in the compressor drive. If no air were present—or decomposed oil or ammonia—then the condenser or head pressure would be determined by the temperature at which liquefaction took place. And the temperature of liquefaction is determined directly by the amount of cooling water showered over the condenser and by the temperature of this water. As an example, it may be said that in acquiring a ton of refrigeration the liquid ammonia in the expansion coils will absorb heat at the rate of 200 heat units (200 B.t.u.) per minute. Additional heat, amounting to about 60 heat units, is added during compression so that the total amount to be removed by the cooling water in the condenser is about 260 heat units per minute per ton of refrigeration. This has to be taken away by the cooling water, which is heated in consequence. By allowing each pound of water to rise 10 deg. F. there would be required $\frac{260}{10}$=26 lbs. (3.1 gal.) of water per minute, or 19½ lbs. (2.34 gal.) when heated 15 deg., or 13 lbs. (1.36 gal) when heated 20 deg. Table 2 shows what can be obtained in practice.

Ammonia Fittings. The steam engineer does not always appreciate the conditions which obtain in refrigeration, perhaps, because of lack of understanding of the cycle employed and through being used to carelessness in steam operation on account of its lesser importance as far as leaks are concerned. The operator should remember that the refrigerating cycle is a closed one, with the compressor in series in order to obtain continuity. With ammonia in particular, although the other refrigerants are hardly less demanding in this respect, the system must be tight. As the pressures

Table 2. Gallons Condenser Water Per Ton Refrigeration

Condenser Pressure, Lb. per Sq. In., Gage	Corresponding Temp. Deg. F.	60 Deg. F. Water: Range, Deg. F.	60 Deg. F. Water: Water Per Ton of Refg'n. Gal. Per Min. Suction Pressure, Lb. Gage 15	20	25	70 Degrees F. Water: Range, Deg. F.	70 Degrees F. Water: Water Per Ton of Refg'n. Gal. Per Min. Suction Pressure, Lb. Gage 15	20	25	80 Degrees F. Water: Range, Deg. F.	80 Degrees F. Water: Water Per Ton of Refg'n. Gal. Per Min. Suction Pressure, Lb. Gage 15	20	25	90 Degrees F. Water: Range, Deg. F.	90 Degrees F. Water: Water Per Ton of Refg'n. Gal. Per Min. Suction Pressure, Lb. Gage 15	20	25
126	75	10	2.90	2.85	2.80												
131	77	12	2.40	2.35	2.30												
136	79	14	2.05	2.00	1.95												
141	81	16	1.85	1.80	1.75	6	4.87	4.80	4.70								
147	83	18	1.65	1.60	1.55	8	3.67	3.60	3.55								
152	85	20	1.47	1.45	1.41	10	2.95	2.90	2.85								
158	87	22	1.35	1.32	1.30	12	2.45	2.40	2.35								
163	89	24	1.25	1.22	1.20	14	2.10	2.05	2.00								
169	91	26	1.15	1.13	1.10	16	1.90	1.85	1.80	6	5.00	4.90	4.80				
175	93	28	1.08	1.05	1.03	18	1.68	1.63	1.60	8	3.75	3.70	3.65				
181	95	30	1.00	0.99	0.97	20	1.50	1.48	1.45	10	3.00	2.95	2.90				
187	97	32	0.95	0.93	0.90	22	1.38	1.35	1.32	12	2.52	2.48	2.43				
194	99	....	....	....	....	24	1.28	1.25	1.22	14	2.13	2.10	2.06				
200	101	....	....	....	....	26	1.17	1.15	1.13	16	1.92	1.85	1.83	6	5.10	5.00	4.90
207	103	....	....	....	....	28	1.10	1.07	1.05	18	1.70	1.67	1.63	8	3.83	3.75	3.70
214	105	....	....	....	....	30	1.02	1.00	0.98	20	1.53	1.50	1.48	10	3.07	3.00	2.95
221	107	....	....	....	....	32	0.95	0.93	0.92	22	1.40	1.37	1.35	12	2.57	2.51	2.47
229	109	....	....	....	....	....	....	....	....	24	1.30	1.28	1.25	14	2.19	2.13	2.10
236	111	....	....	....	....	....	....	....	....	26	1.20	1.16	1.15	16	1.95	1.90	1.88
244	113	....	....	....	....	....	....	....	....	28	1.12	1.08	1.06	18	1.74	1.69	1.65
251	115	....	....	....	....	....	....	....	....	30	1.05	1.02	1.00	20	1.56	1.53	1.50
259	117	....	....	....	....	....	....	....	....	32	1.00	0.98	0.95	22	1.43	1.40	1.37

carried are almost always above the atmosphere the trouble with air leaks into the system is small,* but the piping and fittings must be of a material which will be tight and the



Size Inches	A Inches	B Inches	C Inches	D Inches	All Fittings Have 4 Bolts	
					Diam. Inches	Length Inches
½	3	6	2⅞	1⅞	7/16	2½
¾	3	6	2⅞	1⅞	7/16	2½
1	3⅜	6¾	3½	2⅜	½	2¾
1¼	3⅜	6¾	3½	2⅜	½	2¾
1½	4	8	4	2¾	½	2¾
2	4¾	9½	4½	3 1/16	⅝	3¼
2½	6	12	5½	3⅞	¾	4¼
3	6½	13	6	4⅛	¾	4¼
3½	7	14	6½	4½	¾	4¼
4	7¼	14½	7	5	⅞	4½

Fig. 32. Dimensions for crosses, ells and tees for square flanged fittings. [Arctic Machine Co. practice]

joints must be designed for the kind of work they are called on to perform. As a rule the temperature range that the

*Except through the stuffing box of the piston rod, where air leaks are quite possible.

piping is subjected to is much less than for steam, but steam leaks that would not be noticed would cause considerable loss and complaint when ammonia was in use. In particular there is the trouble with the piston rod, especially if the rod is adjusted out of center. With good erection, and pipe fittings designed for ammonia there is no question but that the ammonia system may be maintained tight.

Design of Ammonia Fittings. Ammonia has the property of escaping piping and fittings which would be absolutely tight for steam. The result is that the material used must be special or it must be of high quality or very thick. The

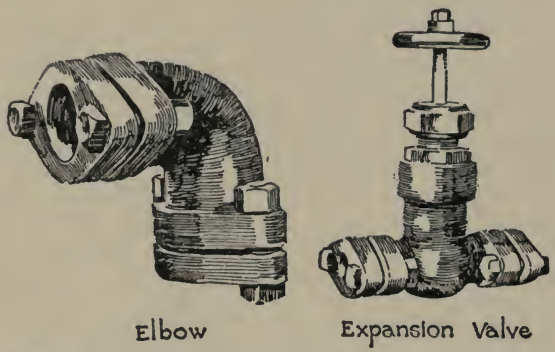


Fig. 33. Ammonia fittings: the elbow and the expansion valve

steam engineer will recollect the difficulty in keeping a vacuum exhaust line tight, and he will then appreciate properly ammonia difficulties. Screwed ammonia fittings may be used and they are made tight in two ways: either by tinning, soldering and (in the case of the larger sizes) by filling the cup of the fitting with solder and calking, or by the use of litharge and glycerine. The first is the preferred method—except possibly where shaking or expansion has the tendency to break the solder—and is generally used in condensers and piping erected in the shops. The second method

can be made strong and tight for a score of years. Flanged joints are designed with the tongue and groove exclusively, either with oval glands or flanges, or with square or round flanges. Both lead and rubber gaskets are used, but in general it may be said that lead must not be used on the discharge side where it is subject to much range of temperature if it is not confined well enough to prevent flowing, an action which is also true in the soldered discharge line under high discharge temperatures. Unfortunately the different manufacturers have not yet standardized fittings except as regards their own products in the matter of the length of run of flanged tees, ells, globe and angle valves, etc. The table (Fig. 32,—taken from the Arctic Company) is an example

Fig. 34. Ammonia fittings; the return bend

Fig. 35. Ammonia fittings; the cross and the pop safety valve

of the present tendency—showing how the different fittings are interchangeable.

The ammonia fitting material is undergoing a steady change. Ordinary cast iron is unsuitable for ammonia, and is supplanted by high grade, usually air furnace, castings, by semi-steel, drop forgings and by machinery steel. Drop forged screwed fittings, expansion and stop valves are becoming popular in spite of the additional cost required in order to drill out and thread. Standard pipe threads are always used. A few typical fittings are shown in Figs. 33 to 37.

With the occasional exception of the suction and discharge bends which are frequently made of cast iron, headers and receivers are now made of steel, usually flame welded, both as regards the longitudinal seam and the heads, and also in respect to the nipples and outlets. Such headers are usually preferable because of the decreased number of joints, and because of their compactness. The finished product is light in weight, moderate in price and (if the work is properly executed) amply strong. The flame welded shapes may be made readily for any design, as it is a matter only of cut-

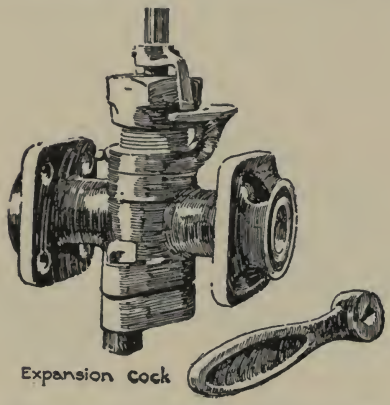


Fig. 36. Ammonia fittings; the expansion cock

Fig. 37. Ammonia fittings; the four bolt flange

ting out the steel plate to the proper shape and rolling to form. (Fig. 38.)

Ammonia Pipe. The kind of pipe to be used is still a question of debate, with the use of the black steel pipe gaining in the race. It has not been our custom to specify seamless pipe as yet, and therefore the pipe used, either wrought iron or steel, is welded from the flat plate. If the welding operation is satisfactorily performed, and the thickness of the walls of the pipe conforms with standard practice, there is no advantage of wrought iron over steel. It appears

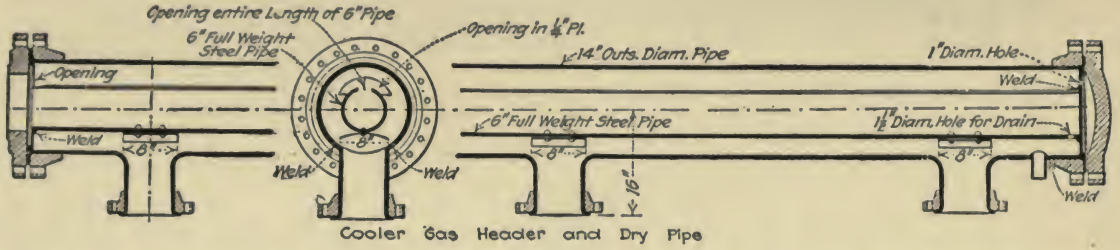


Fig. 38. An example of welded pipe headers. A special design made in the Quincy Market Warehouse and Cold Storage Co.

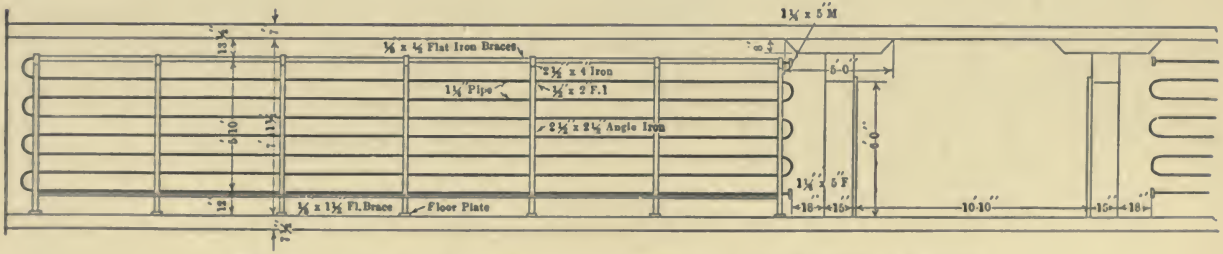


Fig. 39. Continuous piping—Direct expansion. Note absence of fittings

to be simply a matter of getting pipe of good quality in either case. Where corrosion is likely to occur it is becoming the custom merely to use extra heavy pipe, inasmuch as tests do not show any considerable advantage in wrought iron over steel when subjected to corrosion tests. Incidentally corrosion is a very difficult process to prevent, due as it is to electrolysis caused by stray electric currents, and galvanic cell action. Pitting may be formed even by mill scale in the pipe. Both *lap* and *butt* welding of the pipe are permissible, but lap welding is preferable for pipe over 2 in. in normal diameter, and ammonia pipe for smaller diameters than 2 in. should be redrawn from a larger size after welding. In general the lap weld is about 20 per cent stronger than

Fig. 40. Ammonia fittings: one design of an expansion joint

the butt welded pipe. The main requirements in the choice of pipe are careful testing and inspection of all pipe before use.

Without question the modern tendency is towards the reduction of joints in the ammonia line. Welded headers have already been given as an example of this. In addition the expansion line is frequently *welded* and *bent* as shown in Fig. 39, thus requiring a minimum of joints. Expansion joints are usually provided in the larger plants only, and may be of the slip joint design as shown (Fig. 40) or of the flexible U bend. When there is room the latter is far preferable on account of the impossibility of leaks.

The Oil Trap. One of the most serious operating troubles is due to the effect of oil. Most operators feed the lubricating

oil too heavily in the case of the slower rotative speed compressors. The more recent designs of high speed compressors require a very liberal supply of cylinder oil if serious trouble is to be prevented. If oil gets out of the cylinder (and it will) it should be separated from the hot, compressed gas before it enters the condenser. In consequence an oil trap (similar to the steam separator for use on steam engines) is placed in the line, preferably nearest the condenser in order to get the discharged gas as cool as possible before letting it enter the trap. However no oil separator is perfect, no matter how large it is or what design is used, because some of the oil is vaporized by the heat and this vaporized oil will pass into the condenser and will ultimately find its way into the liquid line and the expansion piping. Of course oil in the expansion piping (or any other heat transfer piping) is almost as bad as if it were in boiler tubes. The efficiency of the piping is reduced, and the coils will lose capacity. In consequence it is wise—whenever there is a low point in the expansion piping—to place a special trap for the collection of the oil or other non-volatile liquids. These traps may be connected together and all run into the purifier, or they may be allowed to exhaust into the atmosphere when opened by the operator.

The Regenerator. The regenerator, or purifier, is an arrangement (Fig. 41) which is absolutely necessary for plants of moderate size. The piston rod, being colder than the air in the engine room, has moisture from the air condense on it, and this in time gets into the refrigerating system—being carried in by the movement of the piston. Water will make ammonia "dead," and after a time satisfactory results are impossible. Consequently it is necessary to renew the ammonia charge by pumping out the entire system and recharging, or by using a special device. This latter is an arrangement by which a small amount of liquid ammonia is bled off from the high side, or from the traps, into a special

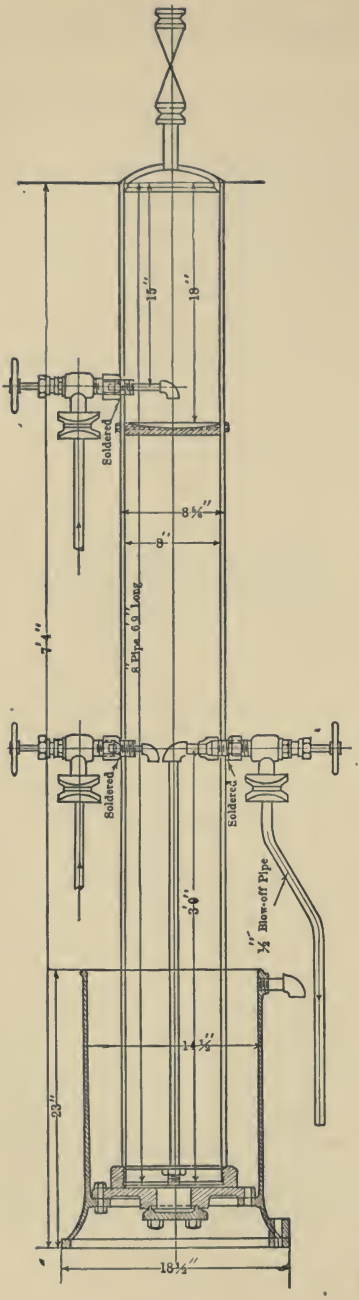


Fig. 41. The Vilter Ammonia regenerator

drum usually arranged for heating slowly by means of hot water or steam coils. Liquid ammonia is bled off regularly and is heated sufficiently in the regenerator to boil off the ammonia which passes to the suction of the compressor. The water and the oil are left behind as a residue.

Every steam engineer knows the test for the level in the boiler by use of the gage cocks, and each engineer has his own idea for the reason for the fact that—when water is allowed to blow out—it appears as wet steam. Apparently,

Fig. 42. Ammonia fittings: the gage glass fitted with four rods for protection against accident

water has flashed into steam on being allowed to escape into the atmosphere. As a matter of fact, the water in the boiler contains more heat energy at boiler pressure than is permissible at atmospheric pressure. As the water reduces in pressure on passing the gage cock, the surplus heat energy vaporizes some of the water, and thus gives the appearance of steam.

The Accumulator. The expansion valve of the refrigerating cycle has been misnamed; it really is only a pressure reducing valve. As the liquid ammonia is fed through the

valve, the pressure drops from, say, 150 lbs. to 15 lbs. or whatever the suction pressure may be in the plant. The heat in a unit weight of liquid ammonia at 60 deg. F. is much greater (because of its temperature) than a similar weight of liquid ammonia at 0 deg. F. In consequence the ammonia in passing the expansion valve has the same effect take place as in the case of the boiler gage cocks, and therefore some ammonia will be vaporized. The amount of ammonia vaporized depends on the temperatures of the liquid on the two sides of the valve and may be less than 10 per cent, but usually is between 10 and 15 per cent. The vapor formed is of no practical value for refrigeration, and the quicker it is gotten rid of the better it will be. Therefore in certain cases a special device, called an accumulator, is used.

The *accumulator* is really a special trap, arranged so that not only does it return to the compressor the gas formed by the process of the reduction of pressure, but it acts also as a separator and insures against liquid returning to the compressor. In one design (Fig. 43) it is seen that there are two parts—the upper acting as a kind of precooler and a suction gas drier, and the lower as a separator for the vapor and liquid and the lowest part as an oil collector. It is noticed that the whole trap is open to the suction pressure and that the liquid ammonia flows to the expansion (refrigerating) coils by gravity. In consequence sufficient head must be allowed to force the liquid into the coils, which are fed from the bottom and are called "flooded." A flooded system of refrigerated coils is not exactly what its term would indicate. As soon as the liquid enters the refrigerating coils, evaporation (and refrigeration) begins and gas is given off. This gas passes on up through the pipe and finally gets into the suction pipe. Thus the coils are all liquid at the beginning, and all or nearly all gaseous at the end (the top) of the coil. The advantage of the flooded system is in the

initial removal of the gas in the accumulator and the easy control of the feed into the cooling coils. The coils are much more efficient in this system, but as a rule it can only be used in ice tanks or special piping where a gravity head may be employed. Especially large pipes must be used in the liquid feed as the head on the system is small at best.

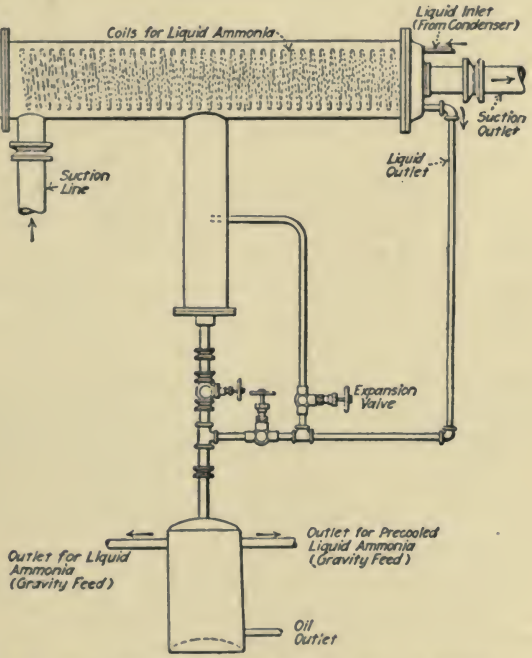


Fig. 43. Ammonia fittings: one type of accumulator using gravity feed of the liquid ammonia into the refrigerating coils

Pump Out. To make repairs or alterations it is required that means be provided to remove the ammonia charge from the part of the system in question and store it in another part. In consequence "pump out" lines are provided. These pipe lines leading back to the compressor need not be so large

as the suction return pipe as it is expected that only small coils or a few stands of pipe will be pumped out at a time, and the pumping process is usually a slow proceeding.

To pump out the high side easily it is customary to have cross connections on the compressor. These are means by which the discharge from the compressor is delivered to the suction pipe and the suction is drawn from the high side. The suction and the discharge stop (king) valves are closed tight, and these bypass valves are placed on each side of the stop valves. In addition, on account of the necessity for freedom in starting electric-driven compressors, a full sized bypass frequently is made between the discharge and the suction pipes. This is really just a "short circuit" so as to reduce to a minimum the power required for starting.

The Scale Trap. On erection the pipe line is carefully blown out to clear of mill scale, etc., but during operation some scale, dirt, foundry sand, etc., will become loosened and will flow back to the compressor. The poppet type of valve is ground to a "line" contact, and other forms are ground to a surface contact. It is clear that impurities of all sorts must be kept out of the cylinder and valves. A leaky valve means much reduced capacity. Therefore a *scale trap* is placed in the suction line near the compressor, where it can be cleaned readily.

The *liquid receiver* (Fig. 44) is another important accessory. It should be large enough to hold the entire ammonia charge for small plants, without flooding the condensers excessively. Both horizontal and vertical receivers are used. The receiver and the discharge line for the compressor should be protected by means of safety valves of the pop-relief type, which may be piped to the suction side as a rule. The suction and the expansion system is frequently protected by relief valves also, piped to the atmosphere or to the blow off mixer where the gas is dissolved in water and flows into the sewer.

Ammonia Stop Valves. Large plants should also be protected in such a manner as to save the ammonia charge should breaks occur. Such protection may be obtained by means of no-pressure stop valves (Fig. 68), or by some form of remote control valve, or by something similar to the automatic stop valve required on each boiler in a battery of boilers.

Carbon Dioxide Fittings. So far nothing has been said of carbon dioxide fittings and accessories, although in general these are not much different from ammonia fittings. Carbon

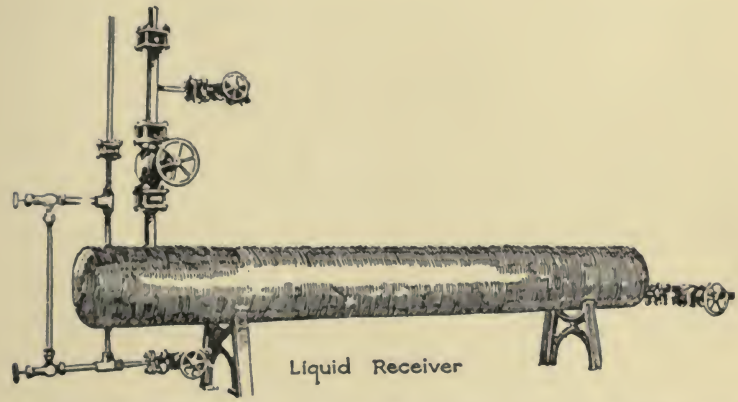


Fig. 44. Ammonia fittings: the liquid receiver and connections. [This is a horizontal type fitted with a liquid gage glass]

dioxide pipe is never heavier than *extra* heavy (even in the case of the 9-in. condenser and brine cooler, which are frequently made of pipe). Even the carbon dioxide compressor frames, bearings, shafts, connecting rods and pins are practically the same as the companion ammonia compressor, inasmuch as the forces applied to these parts are the same in the two. The only exception is that the carbon dioxide machine requires a much heavier flywheel than does the ammonia compressor.

From the foregoing it is clear that refrigerating fittings must be especially designed, first of good thick material, and second with special flanged tongue and grooved joints, or special screwed fittings. Also that, as the system is a closed one, provision must be made for proper care of the refrigerant, for purifying it or caring for it during periods of repair to parts of the system. The piping must be tight, free from low spots unless trapped, and designed with the proper size or the proper surfaces for the work it is called on to do.

CHAPTER IV

Other Refrigerating Systems

The Brine System

Perhaps the operating engineer has never considered that the steam engine or the steam turbine cycle is an indirect one. As a matter of fact, it is an indirect, indirect cycle, because the heat developed by the combustion of the fuel first heated the flue gases, these heated water and formed steam, and the steam was made to generate power in the usual manner. With the internal combustion engine, the oil, or the gas engine, one of these steps (with its losses) is done away with, but the process is still an indirect one.

The Indirect Refrigerating Cycle. So, likewise, in refrigeration an *indirect* system is made use of, which in spite of its inherent losses, yet has so many advantages that these overbalance the disadvantages. It is well to recollect that the so-called direct expansion system—where ammonia is allowed to boil within the pipes in the cold storage rooms, or boxes, there is a remote but great danger latent all the time. The pipe lines may corrode, or the pipe split due to imperfect welding or become broken by accident or by vibration, or the threads or fittings may fail, with the result that some or all of the ammonia in that part of the system may be emptied into the room, causing injury to the goods in storage or even (possibly) fatalities. There have been accidents to liquid receivers where the liquid ammonia has run in streams into the street, boiling all the time and giving off suffocating

fumes along the way. The result is that certain types of service have very nearly discontinued the use of direct expansion of ammonia and confine the cycle within the boundaries of the engine room: that is, service in hotels, apartment houses, district refrigeration and in other places where the refrigerating line cannot be properly cared for.

Besides the loss of life or of commodities should leaks occur, there is the danger in the matter of the initial charge of

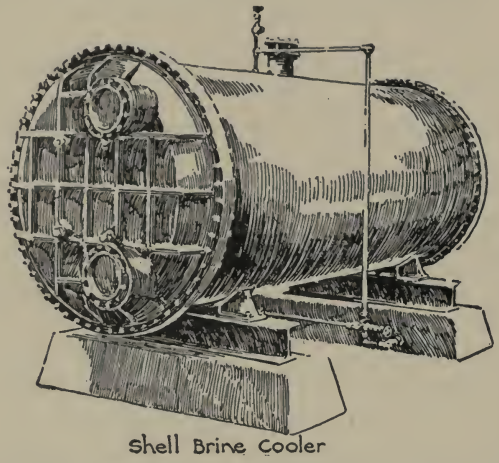


Fig. 45. The shell and tube brine cooler, one of the most successful means of cooling brine

ammonia, which for the direct expansion system may become a very large factor. In a large system of long supply and return lines, and extensive expansion piping, the actual amount of the initial charge has to be very heavy, a fact which results in a constant demand for care and repair to maintain the piping tight at all times. The result is, that, under certain particular conditions, the so-called brine system is best for the refrigerating plant using ammonia as a refrigerant.

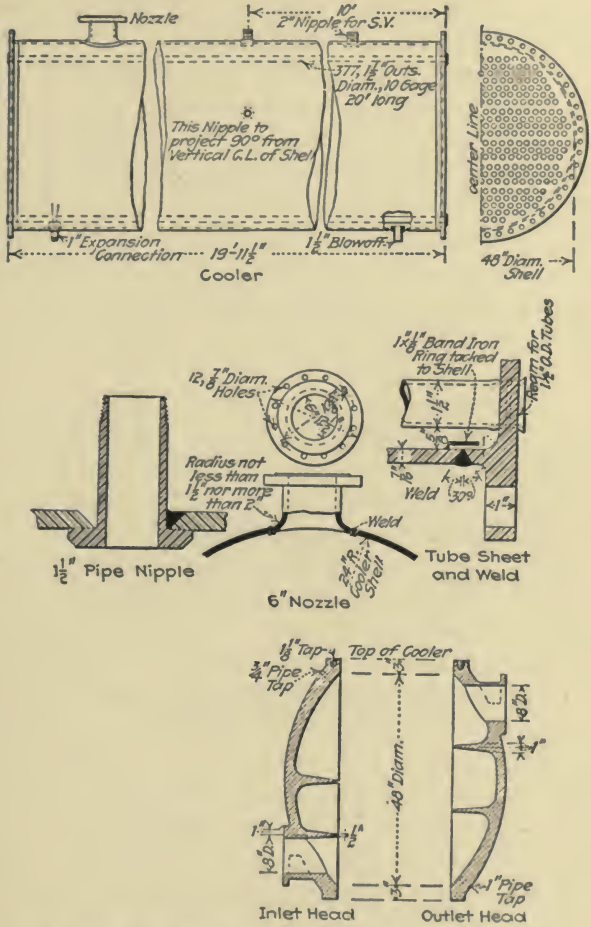


Fig. 46. The shell and tube brine cooler arranged for five passes of the brine

The Brine System. The brine system, then, consists of two distinct circuits similar to the two electric circuits used in alternating current when a transformer is made use of. The ammonia system (or a system using any of the other re-

frigerants, understood by inference) is complete in every detail as brought out in the preceding chapters, but the non-freezing solution is cooled in the double pipe, shell and tube (Figs. 45 and 46) or shell and coil brine cooler. The brine is usually a solution of sodium or calcium chloride of such density as will not freeze at the temperature carried in the cooling system. On account of the lower freezing temperature of calcium brine, it will be found in most service where zero degree F. is required. It is thought also to be less corrosive than "salt" brine, but neither should cause trouble if the solution has a *strength of 1.14,* is *free of* CO_2 and is *slightly alkaline.*

The brine system (Fig. 47) is really an additional unit in the cycle, where the brine is kept cool by boiling ammonia and the cold storage rooms or other refrigerating applications are kept cool by the brine. The ammonia system is self-contained, and the ammonia piping may be kept confined to a single room. The brine, however, must be kept circulating constantly by means of a centrifugal pump, and all exposed piping must be kept well insulated. The brine line need only be full weight steel pipe with ordinary fittings, valves and cocks. The entire piping is laid out in many ways like an ordinary hot water heating plant, which it considerably resembles.

Advantages of Brine System. The great advantage of the brine system, other than those already noted, is in the ability to store up energy. With the direct expansion cycle the compressor must be operating constantly during the live load period. If this is not done, the pipes will soon lose their frost and the room temperature, or the commodities being cooled, will begin to get warmer. With the use of brine this is not necessarily true, as "cold" can be accumulated in the brine storage, and this storage brine may be circulated while the compressor is shut down. In fact, the example can be carried considerably farther in certain kinds

of service that are intermittent, as ice cream making, milk and drinking-water cooling, the chill room in the packing house, or where refrigeration is required suddenly for short periods.

In cases of periodic load the brine tank becomes an advantage. In such a case refrigeration may be stored by cooling a large volume of brine through as large a temperature range as is practical. The compressor may be operated 24 hours of the day, thereby utilizing a much smaller machine than would be required if no brine were used and the compressor capacity had to be capable of the maximum rate of

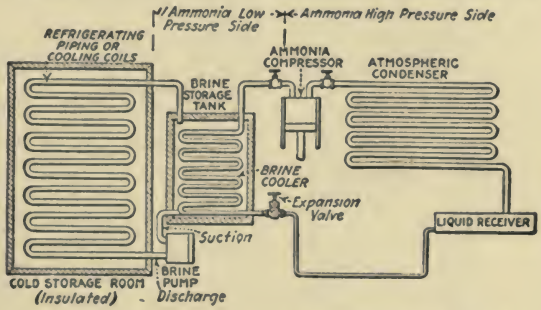


Fig. 47. A diagramatical arrangement showing the brine system of refrigeration

cooling. Sometimes it is an advantage to operate the compressor only 8 to 10 hours per day, and allow the brine pump to operate continuously. It will be seen that by the use of brine the peak load is distributed over the whole 24 hours of the day, whereas the high point of the load may last only three hours. Energy is accumulated in the same sense as in the case of the engine flywheel, or the electric storage battery.

Disadvantages. There are, however, certain disadvantages in the use of brine. The first cost and the operating costs are usually greater than with the direct system. The brine pump has to be operated by independent power. In addition, and more serious than anything else, are the losses incidental to

the use of brine. It will be remembered that the brine is first cooled in the brine cooler, and the ammonia may be required to boil at 16 deg. F. to carry brine at an average temperature of 26 deg. F. Likewise the air to be cooled in the cold storage room could be kept at a temperature of 36 deg. F., thus allowing a 10 degree difference of temperature in each case. The capacity of the compressor would be determined by the temperature of the boiling ammonia, which has to be *only* 26 deg. F. in the case of direct expansion. In all such cases the capacity of the compressor is *roughly* proportional to the abso-

TABLE 3. PROPERTIES OF SODIUM CHLORIDE BRINE

Salt Required		Freezing Point	Degrees Baumé	Specific	Specific
Lbs. Per Gallon	Lbs. Per Cubic Foot	Deg. F.	at 60° F.	Gravity	Heat
0.084	0.63	30.5	1	1.007	0.992
0.169	1.26	29.3	2	1.015	0.984
0.212	1.58	28.6	3	1.019	0.980
0.256	1.92	27.8	3½	1.023	0.976
0.300	2.24	27.1	4	1.026	0.972
0.344	2.57	26.6	4½	1.030	0.968
0.433	3.24	25.2	5½	1.037	0.960
0.523	3.92	23.9	6½	1.045	0.946
0.617	4.63	22.5	7.6	1.053	0.932
0.708	5.3	21.2	8.7	1.061	0.919
0.802	6.0	19.9	9.7	1.068	0.905
0.897	6.7	18.7	10.7	1.076	0.892
1.092	8.2	16.0	12.6	1.091	0.874
1.389	10.4	12.2	15.7	1.115	0.855
1.928	14.4	6.1	20.4	1.155	0.829
2.376	17.8	1.2	24	1.187	0.795
2.488	18.9	0.5	25	1.196	0.783

lute pressure, and in this case the ratio is as 44.12 is to 54.90 or a ratio of 0.8. The compressor operating under the conditions just stated would have a capacity of approximately 80 per cent capacity with the ammonia boiling at 16 as compared with 26 deg. F. In general then the brine system requires a lower back pressure than does the direct expansion, hence demanding a larger compressor, and a greater horse power input per ton of refrigeration. There is also the heat equivalent of the work done in circulating the brine which must be considered and allowed for. For example the work

done in circulating brine by a one horse power brine pump will heat the brine by the amount of the heat equivalent of a horse power—i.e., at the rate of 42.4 B.t.u. per minute or a

TABLE 4. PROPERTIES OF CALCIUM CHLORIDE BRINE
[Bureau of Standards]

Temperature Deg. F	Specific Gravity									
	0.999	1.05	1.10	1.15	1.18	1.20	1.22	1.24	1.26	1.28
70	8.33	8.75	9.17	9.58	9.83	10.00	10.16	10.33	10.49	10.66
60	8.34	8.76	9.18	9.60	9.85	10.01	10.18	10.35	10.52	10.68
50	8.34	8.77	9.19	9.61	9.86	10.03	10.20	10.37	10.54	10.70
40	8.35	8.78	9.21	9.63	9.88	10.05	10.22	10.39	10.56	10.73
30		8.79	9.22	9.64	9.90	10.07	10.24	10.41	10.58	10.75
20			9.23	9.66	9.92	10.09	10.26	10.43	10.60	10.77
10				9.68	9.93	10.11	10.28	10.45	10.62	10.79
0					9.95	10.13	10.30	10.47	10.64	10.81
—10							10.32	10.49	10.66	10.84
—20	Figures are *lbs.* per *Gallon*								10.68	10.86
—30									10.70	10.88
70	62.3	65.5	68.6	71.7	73.5	74.8	76.0	77.3	78.5	79.7
60	62.4	65.6	68.7	71.8	73.7	74.9	76.2	77.4	78.7	79.9
50	62.4	65.6	68.8	71.9	73.8	75.0	76.3	77.6	78.8	80.1
40	62.4	65.7	68.9	72.0	73.9	75.2	76.4	77.7	79.0	80.3
30		65.8	69.0	72.1	74.0	75.3	76.6	77.9	79.2	80.4
20			69.1	72.3	74.2	75.4	76.7	78.0	79.3	80.6
10				72.4	74.3	75.6	76.9	78.2	79.5	80.8
0					74.4	75.7	77.0	78.3	79.6	80.9
—10	*lbs.* per *Cubic Foot*						77.2	78.5	79.8	81.1
—20									80.0	81.3
—30									80.1	81.4
Salt Content Per Cent.	0	6.0	11.6	16.9	19.9	21.9	23.9	25.8	27.7	29.5
Freezing Point	32°	28°	20°	7°	—3°	—10°	—19°	—29°	—40°	—58°
Specific Heat*				0.76	0.725	0.70	0.68	0.664		

*The specific heat is taken for a temperature 15 deg. F. above freezing temperature for each brine concentration.

little more than the equivalent of 1/5 of a ton of refrigeration. The refrigerating machine must neutralize this loss. Sometimes it is noted that the rise of brine temperature during the brine pump operation during the night has been con-

siderable—in proportion to the load on the plant—and frequently the heating effect of the pump is responsible.

Typical Brine Problems. The brine required for a given condition may be calculated easily. The "heat capacity" of a certain volume of brine depends on the product of the density (specific gravity) and its specific heat (see tables 3 and 4) (the heat units required to raise one pound one degree F.). For instance the cooling effect of 100 gallons of brine when raised 4 deg. F. in temperature would be:

4	×	100	×	8⅓	×	1.2	×	0.7	= 2,800 B.t.u.
deg. range		No. gal.		Wt. gal. of water.		Spec. grav. brine.		Spec. heat brine.	

assuming 1.2 for the specific gravity and 0.7 for the specific heat of the brine. The values change with every concentration of brine solution, and the kind of brine used, and reference should be made to the tables of brine solutions.

If it is required to find the amount of brine per minute necessary to provide one ton of refrigeration with a *four* degree range of temperature, the calculation becomes

$$\overbrace{200}^{\text{B.t.u. per ton per min.}} = \text{No. gallons} \times 8\tfrac{1}{3} \times 1.2 \times 0.7 \times 4.$$

$$\text{Therefore—No gallons} = \frac{200}{8\frac{1}{3} \times 1.2 \times 0.7 \times 4} = \frac{200}{28} = 7.14 \text{ gal.}$$

With a four degree range of temperature and the density of brine used in the problem there would be required 7.14 gallons of brine in circulation per minute to provide the carrying capacity of one ton of refrigeration. Should less brine capacity be desired, a larger range of temperature would be required and thereby a lower suction pressure in proportion on the machine.

Automatic Refrigeration

There are many applications of refrigeration which require intermittent work, and especially is this true of motor driven small sized cold storage plants. Being motor driven, usually

of the induction and sometimes of the synchronous type, it is not practical to provide variable speeds to adjust the speed to the load. Also it is desirable where possible to have a machine that is capable of temperature control, one that will be semi-independent of constant care or that will operate by night or day subject to occasional supervision on the part of the engineer.

Holdover Tanks. This desire for an automatic refrigerating machine applies more particularly to the small machine,

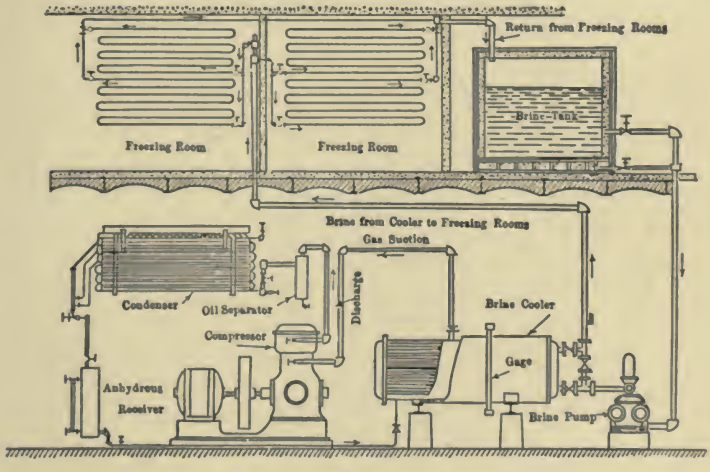


Fig. 48. Typical Brine layout

where the plant does not warrant the cost of additional operators. For very small plants the use of "holdover tanks," or the brine system with large storage tanks and continuous brine circulation, has for some time met with considerable success. However, the holdover tank (see Fig. 6) requires considerable additional headroom, because a tank containing brine and one-third of the total amount of the expansion piping must be arranged at the top of the cold storage room. The submerged piping is expected to provide the necessary re-

frigeration for cooling the brine to an amount sufficient to hold the temperature of the room during the night time.

Likewise the brine storage tank and circulating pump take space and considerable extra power. The compressor, which operates but about one-third of the total time, has to be of sufficient size to carry the day load and store up sufficient refrigeration for the night as well. Also the operation is not so efficient, for the pumping action of the brine pump has a heating effect which alone would heat the brine even if no refrigeration was being done. For example, a one horsepower centrifugal pump would heat the brine at about the same rate that one-fifth of a ton of refrigeration would cool it.

The methods mentioned—holdover tanks and brine tanks—are objectionable on account of the space occupied and the poor efficiency of the operation. The compressor and high pressure side have to be of extra size so as to provide the required tonnage in the time of intermittent operation including as it does not only the day load but the night load as well. Therefore, we have the automatic refrigerating system, which may operate without constant attention the twenty-four hours of the day, or off and on with the demand which is thermally controlled by a thermostat located in the cold storage room. As it may operate full time if absolutely necessary it may be of only two-thirds or one-half the size required of the other designs, which are arranged for part time operation only. Before going into a detailed description of the totally automatic refrigerating machine it will be advisable to review the semi-control systems.

The advent of the small enclosed type of compressor has been a decided advance in the art of refrigeration. It is splash lubricated, and while the crank case remains supplied with oil, it is reasonably free from the dangers incidental to the lack of lubrication. Likewise the main compressor bearings are ring oiled, a method that is reasonably safe as long as the ring or chain revolves freely. There remains, then, the

necessity for the control of the water supply, the excess condenser pressure control and some regulation of the expansion valve (if this last seems justified). In this connection attention is again called to the fact that if the suction pressure is too high no refrigeration will be obtained at all, as the temperature of the boiling ammonia at high pressures may be above the temperature desired. Likewise if the suction pressure is too low the cooling coils will conduct heat better, but they may become choked with a low density gas and the capacity of the compressor will drop to one-third or less of the normal tonnage in consequence of this so-called *specific volume* of the ammonia.

The suction control expansion valve is in principle like the well-known back-pressure valve except that the latter deals only with gas on both sides of the valve while the expansion valve must have liquid on one side and liquid and some 5 to 10 per cent of gas on the other. There is another difference, too, which must be given consideration, namely, that a little liquid ammonia will go a great way in producing refrigeration and the opening must be very small and the valve must be designed for close adjustment. With a head pressure of 150 lbs. on one side and 15 lbs. on the other it is clear that a large amount of liquid could pass through a very small opening, and a pound of liquid ammonia per minute will provide the cooling effect of about two and one-half tons of refrigeration.

The Automatic Expansion Valve. A good example of such an expansion valve is shown in Fig. 49. This is a design using the diaphragm method of control. Liquid ammonia comes in on the right hand side, as shown, and passes through the filter and then to the conical reducing valve, which has an extended stem that presses on a pusher plate. The pusher plate rests on the diaphragm, which is also under the pressure of a control spring. The under side of the diaphragm has suction pressure, and the upper side has a spring

pressure which is regulated by the engineer for the operating pressure desired. Such a device should give close regulation and should be capable of relieving the mind of the operator from all worry about the expansion pipe boiling temperature.

Unless the discharge valve on the compressor is closed or in some inconceivable way the discharge line has become ob-

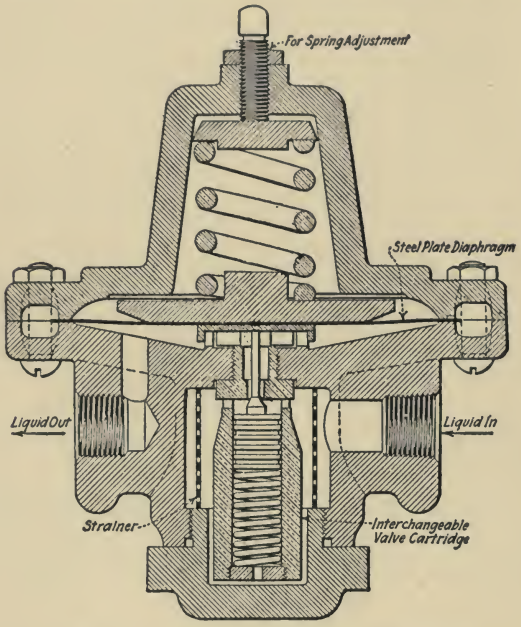


Fig. 49. The automatic diaphragm type of expansion valve—with spring adjustment for control of suction pressure

structed, then the cause of a sudden increase of the head pressure without exception would be the fault of the condensing water used on the condenser. If this condensing water is not turned on or is turned off or fails to function properly then the head pressure will rise. As the ammonia compressor is usually made with small clearances the pressure

that may be obtained under these conditions may be excessive and decidedly dangerous. The result is that some head pressure relief valve must be provided in the automatic system now under discussion.

The Relief Valve. The most elementary of these relief valves are those operating against a spring or a diaphragm and so adjusted as to give movement to a small plunger. This plunger acts on a trip which releases a trigger to which is attached a weighted cord that in falling pulls open the main electric switch (Figs. 50 and 51). Other simple devices are

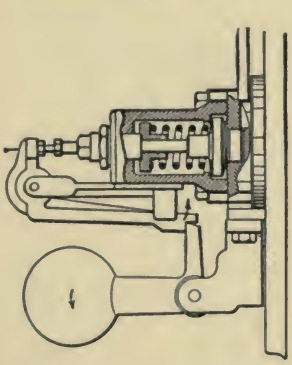


Fig. 50. The excess pressure relief valve

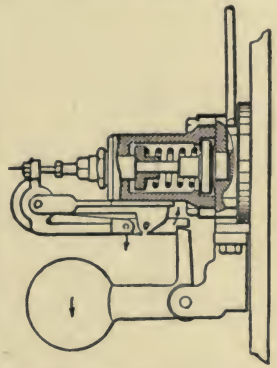


Fig. 51. The low pressure relief valve

on similar lines of design. The relief valve, identical to the steam boiler safety valve, may be placed on the discharge header and piped either to the atmosphere or into the suction header. In the latter case the relief valve becomes simply a bypass valve, but loaded to a certain discharge pressure. Then again, there may be a solenoid device that operates the main switch on the electric circuit. This solenoid is excited by a current that is made when the arm of the pressure gage turns to a certain desired pressure, say 225 lbs., when contact is made and the electromagnet stops the motor. Likewise a device may be used embodying the solenoid principle with that of

the safety valve design. The lift of the valve is made to close the electric circuit, the solenoid is excited and the main switch on the motor circuit is made to open. But all of these designs operate on the head pressure and are planned for shutting down the compressor. While the cooling water is nearly

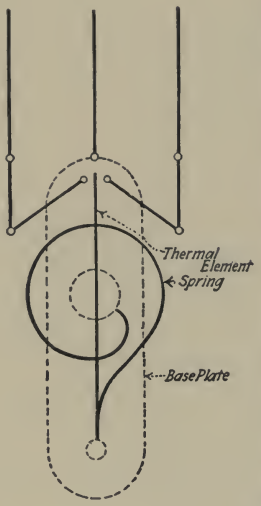


Fig. 52. One design for a thermostat

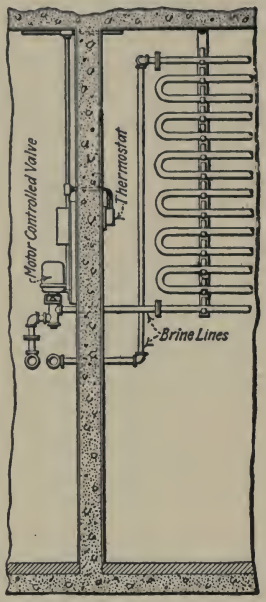


Fig. 53. Thermostat control applied to the brine system

always responsible for excess head pressure, it is not regulated by the controls so far described.

The Water Regulator. The simplest water regulator is one driven by the compressor itself. For the method of control it is necessary to have a pump capable of supplying the condensing water required and subject to some suitable variation of capacity by which more or less volume of water is pumped per stroke of the piston. A variable control of

amounts of water is absolutely required unless, as in some wells, the temperature of the water is constant. A more elaborate valve would be one similar to Fig. 49, which in this case would use the condenser pressure, acting on the lower side of the diaphragm. The rise or fall of this pressure actuates a valve that regulates the amount of water permitted to pass to the condenser, more water being admitted with an increase of condenser pressure.

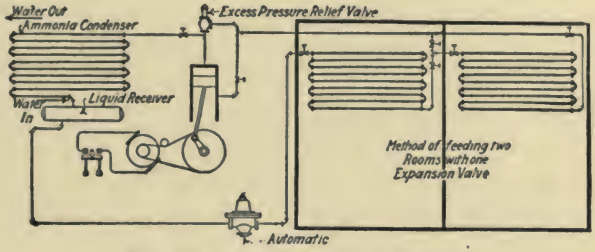


Fig. 54. Semi-automatic system

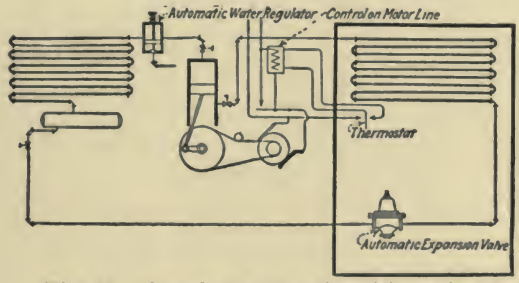


Fig. 55. Complete automatic refrigerating system

To resume, the semi-automatic control system is one that is designed to run under certain prescribed conditions. The suction pressure is to remain constant, the water is adjusted for normal conditions and there is an arrangement for shutting down the machine when the head pressure becomes excessive. The machine is safe under these conditions, and while the compressor is so operated, it is possible to produce useful refrigeration.

The Completely Automatic System. But if the machine stops, it remains stopped until the engineer returns. Also it will continue to operate irrespective of the cold storage temperatures being maintained in the cold storage rooms. The conditions are simply those of *safe* operation during the

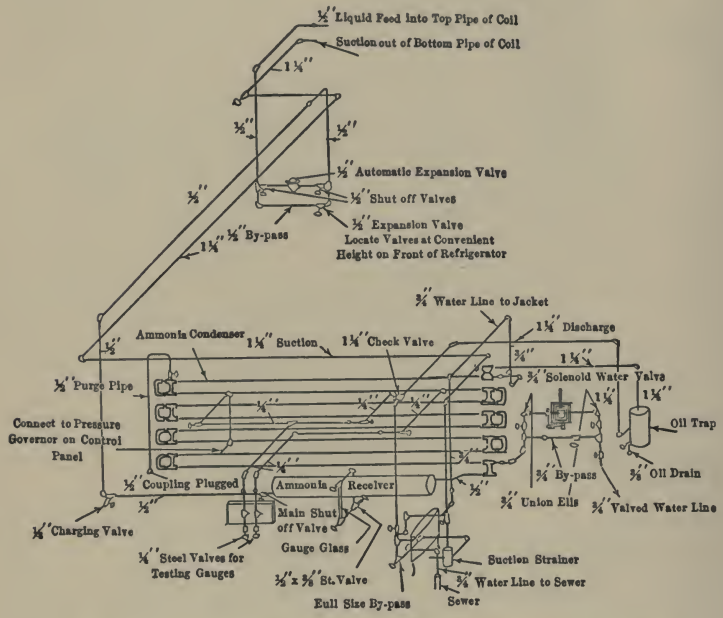


Fig. 56. Isometric showing piping for an automatic system

absence of the engineer in charge. Fig. 54 shows schematically the arrangement of such a refrigerating cycle.

The completely automatic refrigerating system is electrically driven, and so controlled as to require the minimum of attendance for keeping a plant in first class operating condition. If necessary it may operate the twenty-four hours, starting and stopping as required by the temperature of the cold rooms. A *thermostat* is located in the room directly

controlled by the expansion valve, and is the master control of the whole plant.

The Thermostat Control. In general the rise of temperature of the cold storage room expands the elements of the thermostat (Fig. 52) in consequence of which electric contact is made and the electric motor is excited through suitable relays so that the compressor is started. Water is pumped over the condenser by means of a directly connected pump, or the increased head pressure due to starting the compressor actuates the water valve, which opens until a certain condenser pressure is obtained. As the compressor continues pumping, the suction pressure is reduced and the expansion valve is opened, thus allowing liquid ammonia to pass into the expansion coils. As the expansion valve is adjusted for a particular back pressure, it permits a suitable boiling temperature for the ammonia. Operation under these conditions will reduce the cold storage temperature in due course of time and the thermostat will open the circuit and the motor and the compressor will stop. Subsequently a decrease of head pressure will close the water valve (if one is used) and a rise of the boiling pressure (suction pressure) will close the expansion valve. A diagram of such an arrangement is shown in Fig. 55.

Of the automatic features described it is certain that some at least will soon be in common use, especially in the large cities. Safety relief valves are now required, and it is more than likely that remote control valves, or automatic non-return valves (Figs. 35 and 68) on the condenser, receiver and the expansion piping will soon be required also. The main objection to the automatic plant at present is its increased first cost, but it is not unlikely that all plants will be better safeguarded in the future and the disparity in the initial costs will not be so marked. The present boiler plant laws seem to point in that direction.

The Absorption Machine

So far little has been said of the absorption machine, and its system, a type of machine that is least understood in refrigerating circles. However, this machine is important, is adapted to certain kinds of work and will continue to be sold and used for many years. It should be understood, though, that it has its own sphere, and the future will see it more particularly confined to that sphere, and those limitations. For example, it cannot be operated with success without steam. But the high pressure absorption machine is seldom used now, because as a rule the same result may be obtained much more economically by use of low pressure (exhaust) steam or by the use of a steam driven compressor.

Using exhaust steam in the absorption machine, a ton of refrigeration may be obtained for about 30 to 35 lbs. of steam, or about the rate required in certain types of steam driven compressors. In other words we can practically double the tonnage, using the same steam, by the addition of the absorption machine. Is this a case of "getting something for nothing"? The answer is, no. We simply employ another machine, more investment and much more engine room space for the purpose of securing greater economy. But are there any other advantages besides the use of exhaust steam (like exhaust steam heating) for the production of refrigeration with this type of machine?

Advantages of Absorption Machine. Yes, the one other great advantage of the absorption machine lies in the ability to secure low temperature with slight increase of expenditure of power. As has been pointed out a number of times, the compressor gives the maximum economy with the highest possible suction pressure. This is because the tonnage varies directly as the number of pounds of ammonia condensed and later evaporated in the expansion coils. But this number of pounds varies with the so-called specific volume of saturated ammonia, which in turn varies approximately indirectly as

the absolute pressure. At low pressure this specific volume (the volume in cubic feet occupied by one pound) is so great that the standard compressor is frequently expressed as doing nothing but "fanning a vacuum." The standard compressor is unsuited for such service, as it has greatly reduced capacity and tends to overheat the ammonia on discharge. As has been considered already the compound compressor and

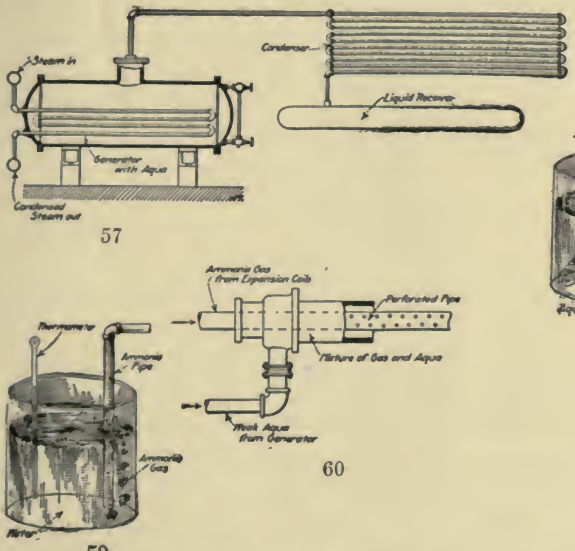


Fig. 57. The high pressure side of the absorption machine
Fig. 58. The principle underlying the generator
Fig. 59. The principle underlying the absorber
Fig. 60. The absorber mixing device in one design

not the standard form of compressor should be used for low temperature refrigeration. But how about the absorption machine? How does it work to secure economy at low refrigerating temperatures?

Operation. The absorption machine is not a displacement pump in the sense of the compressor. The gas is carried over from the low pressure side to the high pressure side in the

form of an ammonia solution at much reduced volume. For example, 5, 10 or 15 lbs. of ammonia solution will carry with it one pound of ammonia. This aqua (solution of ammonia in water) occupies 1/12 to 1/4 of a cubic foot of space per pound of ammonia delivered irrespective of the pressure on the low side, which might be—for the one pound—15, 20 or even more cubic feet. This peculiarity of forming a solution (aqua) is not absolutely liquefaction. It is simply a fact

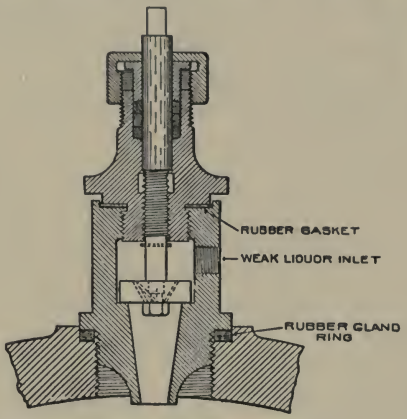


Fig. 61. The mixing nozzle in one design of absorber

that water has the ability to dissolve a large amount of ammonia, say 500 to 1,000 volumes.

The result of the cycle of operations is that the ammonia may be boiled in the refrigerating coils at any temperature desired and the gas formed will be conveyed to the so-called absorber. In the absorber the gas enters solution and the strong aqua formed in the process is pumped back into the "still" again, which is arranged with suitable heating surfaces by means of exhaust steam piping, (Fig. 57). The steam in this piping heats the aqua (so-called strong aqua) in the still or generator and drives off a vapor mixture of steam and ammonia, the steam being almost entirely separated from the

ammonia before condensation occurs in the ordinary type of ammonia condenser. This action in the still is similar to that which takes place when distilling water, refining oil and in other similar industries. The idea is simply that a certain *concentration* of aqua—when heat is applied—will boil ammonia gas (Fig. 58), and the boiling pressure is the vapor pressure in the condenser. Also that boiling will continue until a certain concentration of solution in the still is obtained, unless the temperature of the aqua in the still is increased, in which event boiling will be resumed. As the average temperature in the still in practice is held constant then to have continuity of boiling it is necessary to keep up a steady addition of strong aqua and allow the weak aqua (the strong aqua after boiling away the ammonia gas) to pass out of the still.

The weak aqua from the still is made to go into the absorber where it meets the gas produced by the action of the boiling in the refrigerating coils (Figs. 59, 60, and 61). The latter shows a detailed design of an absorber mixing nozzle. The ammonia gas comes into the absorber and is thoroughly mixed with the liquid weak aqua in a manner similar to the jet condenser for steam; that is, the gas is apparently condensed and carried away. There is, however, no real analogy between the absorber and other steam power plant equipment. The weak aqua does actually condense the gaseous ammonia and then absorbs it, forming thereby a stronger solution or concentration. This strong solution is called the strong aqua, and it leaves the absorber at a low pressure (the pressure in the refrigerating coils), a low temperature but a strong concentration. It is the difference in the concentrations of the weak and the strong aqua which materially affects the economy of the absorption refrigerating machine.

Action in Generator. Suppose we have a generator containing aqua of 26 per cent (26 per cent by weight of ammonia and 76 per cent by weight of water) concentration.

TABLE 5. TEMPERATURES, PRESSURES AND CONCENTRATIONS FOR AQUA AMMONIA

Pressure on Aqua in Pounds Per Square Inch Absolute

*	10	15	20	25	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
2	169.0	189.4	206.5	220.0	231.0	249.4	264.3	277.9	288.8	299.0	307.8	316.2	323.7	330.7								
4	161.0	180.8	197.5	210.5	221.9	240.3	255.0	267.7	278.9	288.6	297.3	305.6	313.4	320.3								
6	153.0	172.3	188.8	201.5	212.4	229.5	244.0	258.9	169.0	278.5	287.4	295.3	203.6	309.6	316.0	322.2						
8	143.5	164.2	180.1	193.1	203.7	221.4	236.0	248.2	259.0	268.5	277.3	285.1	292.5	299.6	305.8	311.6						
10	137.4	154.3	171.1	183.7	194.6	212.3	227.0	239.6	250.5	259.5	267.9	275.8	282.9	289.4	295.7	301.8	307.4	312.6				
12	129.0	147.8	163.1	175.1	185.5	203.3	217.2	229.2	240.0	249.4	258.1	265.6	272.7	279.7	285.6	291.5	297.0	302.2				
14	120.5	140.1	155.5	167.4	177.6	195.2	209.1	220.8	231.2	240.5	248.9	256.1	263.0	269.7	276.0	281.8	287.3	292.4	297.1	302.0		
16	113.0	133.0	147.4	159.3	169.4	187.1	200.5	212.0	222.3	231.6	239.9	247.3	254.0	260.4	266.4	272.2	277.5	282.8	287.3	292.0		
18	107.5	124.9	139.4	151.4	161.2	178.3	192.0	203.1	213.5	222.6	230.9	238.0	244.7	251.0	257.0	262.5	268.0	273.0	277.6	282.4	287.0	291.7
20	99.0	117.3	132.0	143.6	153.8	170.2	183.6	195.4	205.0	213.8	221.8	229.1	235.9	242.0	247.9	253.5	258.6	263.6	268.4	273.3	278.0	282.7
22	92.0	109.9	124.6	139.9	146.0	162.7	175.9	187.0	196.9	205.6	213.5	220.6	227.1	233.4	239.3	244.7	250.0	254.9	259.3	263.9	268.5	273.0
24	86.0	103.0	117.0	129.5	139.2	155.2	168.7	180.0	190.0	198.4	206.0	213.0	219.2	225.5	231.3	236.7	241.9	246.7	251.0	255.5	260.0	264.5
26	79.2	96.3	110.6	122.1	131.9	148.2	161.0	172.1	182.0	190.7	198.2	205.1	211.5	217.6	223.4	228.7	233.7	238.4	243.0	247.2	251.5	255.8
28	73.4	90.6	103.6	115.3	125.0	140.6	154.0	165.2	174.7	182.8	190.3	197.1	203.1	209.3	215.0	220.4	225.5	230.3	234.5	238.9	243.2	247.5
30	66.5	83.8	97.0	108.5	117.8	133.5	146.1	157.0	166.4	174.9	182.7	189.4	195.6	201.3	206.9	212.2	217.3	222.0	226.7	231.1	235.1	239.3
32	61.0	76.9	90.0	101.0	111.0	127.1	139.4	149.7	158.8	167.3	175.0	182.0	188.2	194.0	198.4	204.5	209.4	213.8	217.7	222.0	226.4	230.8
34	55.0	71.6	84.0	94.8	104.5	120.0	132.0	142.4	151.9	160.2	167.8	174.6	180.9	186.6	191.9	197.0	201.7	206.1	210.0	214.2	218.3	322.5
36	49.0	65.5	78.5	88.8	98.1	113.5	125.8	136.0	145.3	153.3	160.7	167.3	173.5	179.3	184.7	189.7	194.5	199.1	202.9	206.8	210.6	214.4
38	43.0	60.0	72.5	83.2	91.8	106.8	119.4	130.0	138.7	146.7	153.8	160.3	166.4	172.2	177.5	182.5	187.3	191.7	195.6	199.3	203.0	206.9
40	37.0	53.8	66.0	77.3	86.0	101.0	112.9	123.4	132.3	140.0	147.0	153.4	159.4	165.0	170.5	175.5	180.1	184.6	188.9	192.8	196.5	200.1
42	31.6	48.7	61.0	71.5	79.4	94.0	106.3	116.4	124.9	132.5	139.8	146.4	152.7	158.2	163.5	168.5	173.3	177.7	181.4	185.3	189.2	193.0
44	27.5	43.2	55.3	65.3	73.6	87.8	100.1	110.3	118.8	126.3	133.4	139.9	146.0	151.5	156.7	161.6	166.3	170.8	174.3	178.2	182.0	185.0
46	21.2	37.6	49.7	59.6	68.1	82.0	94.0	104.2	112.8	120.1	126.9	133.4	139.5	145.0	150.2	155.0	159.4	163.8	167.5	171.3	175.2	179.9
48	16.0	32.0	44.4	54.2	62.7	76.7	88.3	98.5	106.7	114.0	120.6	127.0	132.9	138.4	143.7	148.6	153.3	157.5	161.2	164.9	168.6	172.5
50	11.0	26.8	38.9	48.4	56.8	70.9	82.6	92.6	100.8	108.3	114.9	121.0	126.9	132.4	137.2	142.0	146.4	150.7	154.9	158.7	162.4	166.0

*Concentrations.

Suppose that a condenser pressure of 161 lbs. gage is carried—this is determined by the amount and the temperature of the cooling water circulated in the condenser—such a solution at this temperature and condenser pressure will boil at 245.5 deg. F. This is similar to the action of heat on pure water. We say ordinarily that water in an open dish will boil at 212 deg. F.; also that the temperature of the water (and steam) in a boiler at 125 lbs. gage is 353.1 deg. F. and the temperature of steam under a vacuum of 26 in. is about 125.0 deg. F. In other words, when speaking of saturated steam, there is a definite, unvarying pressure for each particular temperature. This is also true for anhydrous ammonia, but aqua ammonia is different, because there is water present, varying in amount from 70 to 80 per cent. But if we have a certain concentration and a certain pressure, then the solution has a maximum concentration at a certain particular per cent. That means that at 26 per cent under the pressure of 161 lbs. and a temperature of 245.5 deg. F. there could not be any more ammonia dissolved. In other words a saturated concentration of 26 per cent and a condenser pressure of 161 lbs. will have a temperature of 245.5 degrees. But more important than this, if the pressure is 161 lbs. and a temperature of 245.5 degrees is maintained, a concentration greater than 26 per cent will cause some ammonia to be driven off as a gas. If the solution was 27 per cent, 28 per cent or 30 per cent, then while the temperature and the pressure were maintained constant a boiling action would continue until the remaining solution had a strength of 26 per cent. This is true of any solution of ammonia and water; that is, there is a fixed relationship of the pressure, the temperature and the concentration at saturation, and if any two are known, the third may be found, as will be seen from Table 5.

Such being the theory of the action of the generator, let us see how the action is accomplished. In the generator steam

coils are provided, so arranged as to supply the heating surface necessary to drive off the ammonia gas and the steam. This is somewhat similar in manner to the case of the steam boiler. In the water tube boiler we usually say that 10 sq. ft. of heating surface is required per boiler horse power. In the generator 14, 15 or 16 sq. ft. of heating surface is used per ton of refrigeration. Of course, in the generator we could use very hot steam in the heating coils—say steam at 75 lbs. or 100 lbs. pressure—and in consequence of such intense heat (comparatively) the heating surface could be cut down to 12 or 10 or even 8 sq. ft. In the case cited it might be desirable to boil the solution down to 22 per cent—requiring a temperature of the aqua of 262.1 deg. F.—and as a mean concentration of 26 per cent would be used, then 30 per cent concentration could be looked for in the strong aqua. The steam pressure must be such as to give a sufficient temperature difference between the steam in the coils and the aqua. For example, 262.1 + 10 = 272.1 or a 28-lb. steam pressure is required.

The action then of the generator is to boil off a vapor which is slightly superheated. This is obtained by having a temperature sufficiently great in the steam coils, determined by means of the values given in the table. Increasing the steam pressure or increasing the concentration will stimulate the boiling action, but the easiest point of control for the operator is the steam pressure. If capacity is not being obtained a rise of steam pressure will give hotter steam and therefore a greater temperature difference and a greater heat transfer. The result will be that more steam will be condensed in the coils and more ammonia gas will be distilled out of the generator.

It is readily seen from the foregoing that one of two things in the generator may be wrong if the plant is found not to condense the ammonia properly—either there is too weak a concentration in the generator or the steam pressure is too low.

In either case charging more anhydrous into the system or increasing the steam pressure will give the same results; namely, having enough difference between the condensing temperature of the steam and the boiling temperature of the aqua to enable sufficient heat to pass through the heating surface of the steam coils. All of this may be seen from the table. For instance, suppose it is found that ammonia is not being condensed fast enough in the condenser, though the latter as well as the rest of the piping between the generator and the condenser is working properly. Suppose that the steam temperature is 250 deg. F. and the condenser pressure is 161 lbs. It is possible to take a sample of the weak aqua and test it for its concentration. This, however, is not necessary if we have a means of measuring the temperature of the aqua as it leaves the bottom of the generator. Suppose the temperature is 245 deg. F. Referring to the table, the concentration is found to be only 26 per cent—the supposition being that the aqua leaves the generator in a *saturated* condition. If this concentration had been 28 per cent—with a saturated temperature of 237.1 deg. F.—the saturation *temperature* would have been found to be lower and for the same steam temperature there would have been more degrees difference of temperature on the two sides of the heating coils and therefore a greater heat transfer resulting in a greater number of pounds of ammonia evaporated. The same result necessarily would have occurred with an increase of steam pressure.

Action in Absorber. In the absorber the vapor tension is determined by the temperature and concentration of the strong aqua in a manner similar to that used in the case of the generator. For example, if the strong aqua is 90 deg. F. and 32 per cent aqua is being carried, then the vapor tension is (from the tables) 20 lbs. abs. or 5 lbs. gage. But this 5 lbs. gage is possible only when the absorber is purged out. A new plant is likely to have a large amount of air in the

system, and this will slowly accumulate in the absorber. If air collects, which will itself exert a pressure of 3 lbs. per square inch, then the total pressure in the absorber will be 8 lbs. The boiling pressure of the ammonia in the expansion coils is *not* the absorber pressure. It tends to become this pressure, but in fact there is usually considerable "wire-drawing" between the two pieces of apparatus, and the result is that the pressures of the brine cooler or expansion coils are usually from 2 to 5 lbs. in excess of the absorber pressure. This loss of pressure may be eliminated by making the pipe line connections and gas headers of liberal size. The inert gas may be removed in the same way that is used in the ammonia condenser.

Aqua Concentrations. The concentrations carried in the system are of great importance as regards both the economy

TABLE 6. POUNDS OF STRONG AQUA REQUIRED PER POUND OF AMMONIA EVAPORATED

Concentration	Concentration—Strong Aqua							
Weak Aqua	20%	22%	24%	26%	28%	30%	32%	34%
18%	41	20½	13⅔	10¼	8.2	6.84	5.86	5.12
20%	...	40	20	13⅓	10	8.0	6.77	5.71
22%	...	....	39	19½	13	8.75	7.8	6.5
24%	...	....	....	38	19	12⅔	9½	7⅔
26%	...	....	....	....	37	18½	13⅓	9.25
28%	...	....	....	....	....	36	18	13.0

of operation and the capacity of the plant. The reason for this statement may be seen on examining the action of the strong aqua pump. This pump has an action similar to the boiler feed pump which takes the feed from an open feed water heater. The aqua pumped is only a *carrier;* a pound of the aqua sometimes having but little more ammonia within it than did the weak aqua on entering the absorber. This strong aqua *carries* back to the generator the ammonia gas absorbed in the absorber. Therefore if it takes 13 1/3 lbs. of strong aqua under certain conditions to carry back one pound of ammonia, a drop of concentration from 32 to 30 per cent would necessitate pumping 18½ lbs. of aqua, or one-half more

than before. Thus, evidently, it is important to keep the strong aqua as strong as possible, and for satisfactory operation of the plant a means of getting the strong aqua temperature as it leaves the absorber should be provided. If no inert gases are present, then the concentration may be obtained from the aqua tables as noted. The action of concentrations on the pounds of aqua required to be pumped are shown in Table 6.

Generally speaking, it is wise to keep the difference in the concentrations as great as possible. The reason for this is that the weak aqua has to be cooled by cooling water before being admitted to the absorber, and also the strong aqua has to be returned to the generator at a temperature much below the temperature of the aqua there. This latter action is as bad for the generator as far as economy is concerned as pumping cold water into a steam boiler. For before the ammonia can be distilled the aqua must be heated up to the temperature of the solution in the still. If a very little aqua is circulated (large differences of aqua concentrations being used) then this loss is small, but if a large amount of aqua is pumped the loss is appreciable and should be corrected.

The *exchanger* is another necessary piece of apparatus in the absorption system. It has been noted that the strong aqua has to be cool to absorb the ammonia gas, and should be hot before it returns to the generator. Likewise the weak aqua is hot on leaving the generator and must be cool before entering the absorber in order to be able to absorb the gas. In consequence the exchange is used. This is simply a counter flow apparatus that allows the heat of the weak aqua to pass by metal contact to the strong aqua. In addition a weak aqua cooler is used to precool the weak aqua a little more with water. Finally the vapors leaving the generator consist of a mixture of steam and ammonia gas superheated over a hundred degrees. To remove some of this superheat and particularly to condense the steam so that it

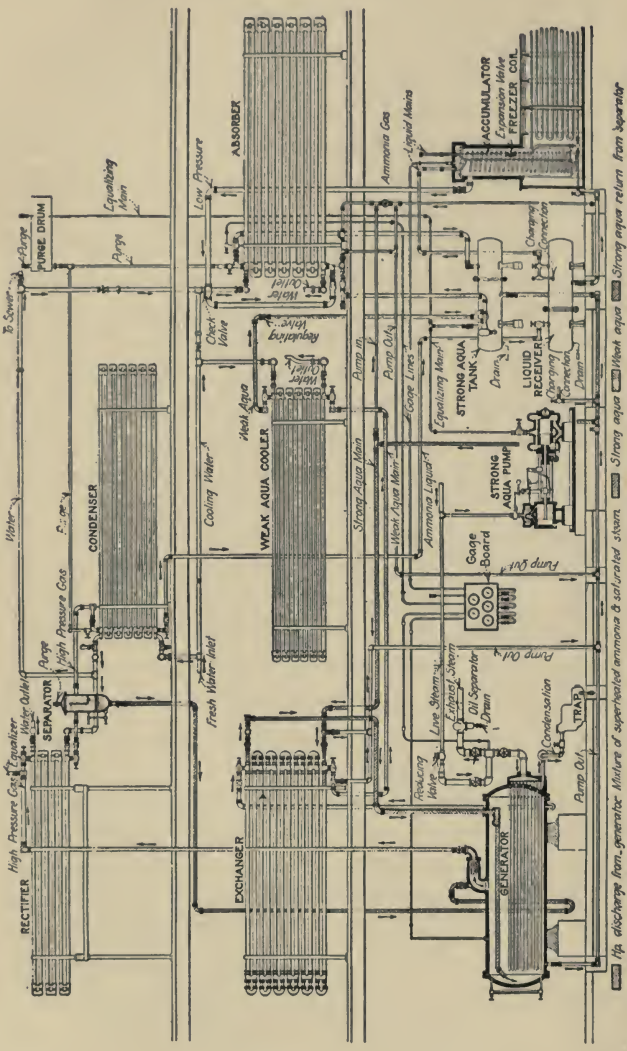


Fig. 62. The complete absorption machine designed for can ice making

will not get into the condenser the so-called rectifier is used. This rectifier or dehydrator is water cooled and is designed to cool the gas to within 15 to 20 deg. F. of the condensing temperature. During this cooling almost all of the steam is condensed and returns through suitable traps to the generator in the form of a very strong aqua. The cooled ammonia gas, with a trace of water remaining in it, goes now to the condenser. The absorption system is shown in Fig. 62.

It may be said that the absorption machine is somewhat like a steam heating system except for the addition of certain special pieces of apparatus. The plant has a single moving machine, namely, the strong aqua pump (provided the cooling and condensing water does not require pumping). The plant is also like a distilling system and when understood, will give entire satisfaction. However, the action in the generator and the absorber must be understood, and the temperatures in the system as well as the concentrations carried must be maintained at the proper values to get good results. The rectifier must not cool the ammonia too much or it will act as a real condenser and the ammonia will be trapped back to the generator. Sometimes it is possible to economize in the use of the cooling water, especially if it is cold, by sending it through several parts of the cycle in succession, but such schemes are not common. Usually there is required some two or three times the amount of water needed by the standard compressor.

Combination Plants. The particular advantages in the use of the absorption machine lie in the relative ease in securing low temperatures and in the ready use of exhaust steam for power purposes. Where high pressure steam has already been used and is available as exhaust steam in sufficient quantities at a nominal price (or as a waste product) the absorption machine may be used advantageously. The economy of operation must be sufficient to warrant the installation. It is becoming quite popular to install *combination* plants, using

the exhaust steam from steam driven compressors and auxiliary apparatus to operate the absorption machine. In general it can be said that small steam engines should not be operated condensing under vacuum. In fact no simple engine should be so operated, because the losses due to cylinder condensation make such operation a loss instead of an advantage. But, where exhaust steam cannot be used for useful heating or drying purposes, the absorption machine makes a satisfactory addition in a combination plant.

CHAPTER V

Erection and Operation

It is not possible to give more than fundamental ideas in a brief discussion such as this, and especially is this true of a subject of the importance of erection and operation. As detail design is so different with different machines it will be possible to make only general statements such as will be true for all makes.

The Principal Points on Erection. The essentials in refrigeration are to make the system *ammonia-tight,* the valves so that they will open properly with the least excess pressure possible and close properly and promptly. Of course the valves must be ground to a tight seat. The compressor must be erected so that the reciprocating parts are central, for it is impossible to maintain tight rod packing otherwise. Adjustments should be made so as to give more clearance on the head end for the double acting machine, because of the effect of friction on the length of the piston rod. In this connection it will be remembered that the operator can easily stall the compressor by tightening on the stuffing box packing. The enclosed type of compressor, having no stuffing box, but having the entire connecting rod exposed to the temperature of the suction gas, which is colder than the temperature of the air prevailing during erection, should have no clearance. In this case the rod will *shorten* with decrease of temperature as soon as operation begins.

The Piping. Contrary to the steam engine cycle the refrigerating system is a closed one and the plant will retain all the impurities that are left in it during erection. In consequence care should be taken to make the inside as free from pipe threads, mill scale and dirt of all sorts as possible. Before erection all pipe should be *cleaned,* at least as far as hammering on one end. After testing with compressed air for leaks it is usual to open each coil or section of pipe directly to the atmosphere to blow the loose stuff out of the system.

The piping should be carefully erected. Ammonia is very difficult to keep tight, and special provision is always made to stop leaks. This includes special (ammonia) fittings of dense, close-grained cast iron, steel castings and drop forgings, these last drilled out and threaded from the solid metal. The pipe used should be of wrought iron or of steel, either butt or lap welded in the smaller sizes, but only lap welded in sizes larger than 2 in. diameter. When steel pipe is used it is better to specify extra heavy rather than full weight, especially if the plant is troubled with factors that cause corrosion, such as heavy galvanic action or stray electric currents.

Joints should be made up with litharge and glycerine (always mixing up a little at a time as needed) or by use of the sweated joint, this last being the preferred method if the pipe line is not subject to an action which has a tendency to cause the solder to break. The work done at the shop, such as condensers and other piping put up in stands, is preferably sweated—by tinning and then making a shrink fit with enough solder to fill the screw threads and the cup at the back of the fittings. The piping should be securely fastened and arranged without low spots where water, oil or mud will accumulate. Where low spots are necessary the piping should be arranged to drain to this spot and a trap

should be placed there. It will assist in keeping the system free from impurities and will be handy during overhauling.

Traps. The trap for small plants may discharge into the atmosphere, but for larger plants it should be connected with other traps feeding into the regenerator. The regenerator should always be used in plants of any appreciable size. It is the only way that we have of keeping the ammonia "alive" and in good operating condition without the necessity of removing the charge and returning it to the manufacturers of ammonia for purification. A suction-line trap should be placed in the pipe line near the compressor to prevent the passage of scale and other impurity into the compressor. This trap acts in a manner similar to the steam engine separator and will assist in preventing slugs of ammonia as a liquid entering the compressor during operation.

A satisfactory trap in the suction-line is one of the most important single pieces of apparatus, and it should be made plenty large enough, and provision should be made to keep it properly drained. The greatest operating danger in the refrigerating plant is that of slugs of liquid returning from the expansion coils to the compressor. Ammonia compressors are not designed to utilize relief valves in the cylinders, and these are a questionable advantage even with steam cylinders. Therefore, for safety, liquid must be prevented from entering the cylinder, and the suction trap must be capable of taking care of any liquid that may come over with the gas. The liquid ammonia caught in the trap must be passed into the coils again (if there is head enough); otherwise it must be pumped back or allowed to go to the regenerator. The trap must be cleaned frequently under any circumstances.

The Oil Separator. It is a mistake to place the oil (discharge) trap too near the compressor. With the highest grades of lubricants some oil will volatilize with the temperature rise during compression, and unless this is con-

densed again *before* it reaches the oil trap it will pass on to the condenser. The discharge line, being uncovered, will quickly lose its superheat and there will be more chance of complete removal of the cylinder oil if the trap is placed nearest the condenser. This trap is likewise similar in action to the steam separator and is preferably made large so that the liquids may have a chance to separate out by gravity.

Loss of Capacity. It is now universally conceded that the most economical manner of operation is to bring the ammonia back to the compressor in a dry, saturated condition. Too much superheat of the return gas cuts down the capacity of the machine, and too wet a condition (wet compression) of the gas allows excessive re-expansion of the clearance volume as well as inefficient use of the liquid ammonia. The temperature of the gas entering and leaving the compressor is important and every machine should be provided with thermometers in the suction and the discharge lines. By means of thermometers the operator can tell whether the suction gas is superheated or wet by reading both thermometers and comparing the readings found with the temperature corresponding to that of the suction pressure. It should be noted that if too much liquid is returning with the gas the temperature of the discharge will be low—possibly cold to the hand. These readings are similar to those in the steam power plants, the CO_2 apparatus, the steam flow meter, the boiler draft gage, etc., and are a ready means of showing changes in operating conditions, as just mentioned. The old method of trying to operate by placing one's hand on the pipe should be avoided, as it has been found to lead to very bad results.

Reviewing the operating conditions which cause *decreased capacity* of plants, increased power input or inability to secure the required temperature it will be found that the main troubles are valve action, ammonia contamination with

inert gases in the condenser and water in the liquid receiver and the expansion coils, and scale, oil, mud and other pipe troubles in the condenser and the expansion coils.

The operating engineer who is accustomed to the operation of steam machinery will find it necessary to make due allowance for the refrigerating cycle. The greatest advantage of steam machinery is that it will always work. In fact, this characteristic has been one reason for its continued employment, frequently, but refrigerating machinery has to be operated just right. Leaky flanges, fittings, expansion joints, stuffing boxes, valve rods, etc., might be all right with steam, but all wrong with ammonia. Sooty and scaly boiler tubes might "get by" but similar contamination in the refrigerating cycle will cause much reduced capacity if operation is possible at all. The fact is that instead of several thousand degrees difference of temperature on the two sides of the pipes (as in the case of the steam boiler), the expansion pipes have only 10 or 15 degrees difference of temperature. Then again the steam engine can be operated with 1/16 to 3/32 of an inch out of round, but failure would be the only result of such conditions with the use of ammonia. The capacity of the compressor is determined from the amount of gas actually *pumped* into the condenser, and leaks through suction and discharge valves or past the piston mean that much reduction in capacity.

In consequence the equipment must be erected or overhauled so that it will be *sound* and *tight*. The valves must be gas tight, and the action must be such as will allow the least "slip" possible. After operation for a short time the valves should work freely, and must be perfectly smooth. On erecting it is wise to operate the compressor on air for a few days, with discharge open to the atmosphere and with plenty of oil, so as to be sure that the adjustments are right. Another trouble may be expected—that due to sand or scale getting under the valves. This trouble is due to the loosening

of sand from the unfinished surfaces of castings, or of mill scale in pipe or of dirt in the system not blown out during erection.

Inert Gases. Inert gases in the condenser are not only bad because of the extra pressure which the compressor has to pump against, but also because of the decreased efficiency of the condenser containing them. As already explained, air, nitrogen and hydrogen in the condenser act as an insulator on the pipes and it is possible under these conditions for parts of the pipes in the stand to be out of commission. Inert gases are mainly due to the presence of air and decomposed lubricating oil. There is little decomposition of the

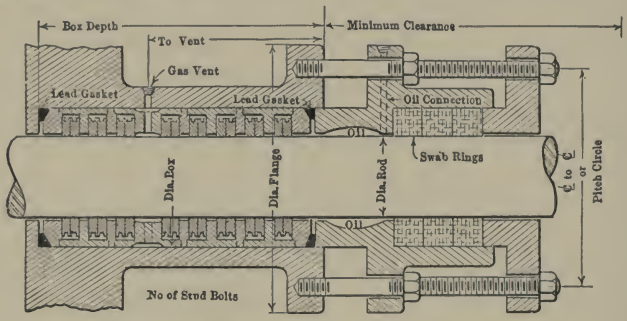


Fig. 63. Metallic packing (France Co.) for piston rods

ammonia at the usual operating conditions. Care should be taken never to operate at a vacuum, unless the exceptional condition prevails of necessity of very low boiling temperatures of the ammonia. The trouble with operating under a vacuum is that air will always get into the cylinder through the stuffing boxes, even though the packing is sufficiently tight to prevent the ammonia escaping into the atmosphere when the suction pressure is greater than the atmosphere. *Metallic packing* is frequently worth the increased cost, because of reduced friction and reduced ammonia losses. (Fig. 63.) *Lubricating oil* should be selected for the two extremes of temperature, so that it will not freeze at the suction tem-

perature or disintegrate at the temperature of the discharge under the most severe operating conditions. Incidentally the maximum temperature of the discharge gas is obtained when the suction pressure is the lowest and the condenser pressure the highest. The condenser pressure should be frequently compared with the ammonia tables, using a temperature a few degrees higher than that of the exit condensing water. Any marked difference between these two pressures indicates that non-condensable gas is present in the condenser, which

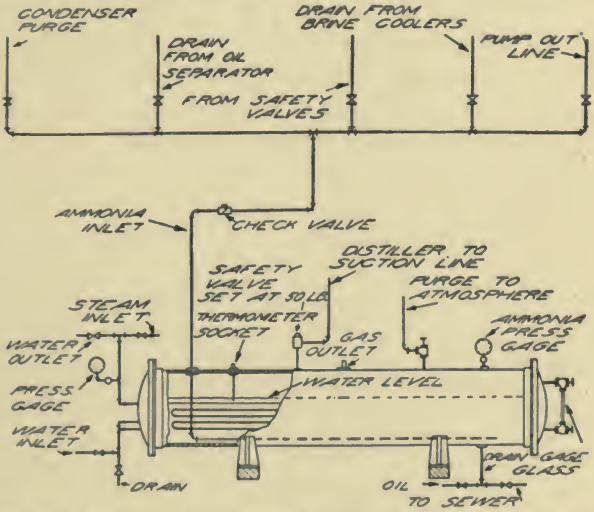


Fig. 64. A special regenerator device for purge connections

may be removed only by purging. But the reader is warned against too frequent purging, as it has been proved that the greater part of the ammonia lost in some plants has been due to this cause. It is impossible to purge without allowing a relatively large amount of ammonia to pass out of the system with the inert gas. All large plants have devices to reduce this loss to a minimum, by purging into water and, similarly to the action of the generator of the absorption machine (Fig. 64), reclaiming this ammonia by heating, or by the

use of the refrigerated purge tank. The latter is simply a device to cool the gases being purged by the means of refrigerated coils—the idea being to condense the ammonia out of the permanent gases in a manner comparable with the action of the rectifier of the absorption machine.

The liquid receiver should always have a *gage glass* with automatic valves that will shut off the connection should the gage glass break. The liquid receiver level is always changing, and if more than one temperature is being carried continual trouble will be encountered unless the operator can tell how much liquid he has back of the expansion valve. If the receiver runs empty then liquid and uncondensed gas will flow into the expansion pipes, thereby choking the coils and reducing the refrigerating ability of the system.

The Stick Test. In this connection, as well as in any expansion line test, it is convenient to use the "stick test." This test, for boiling temperatures of ammonia below 15 deg. F., is to scrape away the frost on the pipe and to apply a moist finger. If the proper temperature is being carried within the pipe the finger will stick and it will require quite a pull to release it. If the finger does not stick it shows that the boiling temperature is not that which is expected and the trouble must be found and rectified.

Sometimes the operator will find the *discharge temperature* of the gas leaving the compressor a means of locating trouble. As no up-to-date engineer now advocates the use of wet compression (unless the compressor does not have a water jacket) the presence of a cold discharge gas pipe means that considerable liquid is returning to the machine. This means that one of the expansion valves is too open, that (possibly) there is a leaky expansion valve, or that the coils are in a flooded condition. By applying the stick test to the different suction return lines it is possible frequently to locate the trouble quickly.

Oil. The almost daily care to overcome the trouble due to oil in the system is comparable with the use of the surface blow and the daily blowing off of the steam boiler through the blow off valves. The engineer will be aware certainly, that solids in the feed water, and possibly oil in the condensate will have to be removed from the boiler by means of the blow off valves or by washing out the boiler. In the refrigerating system there is an accumulation of oil inside the pipe (condenser, receiver, traps and expansion piping) and frost and scale on the outside of the expansion piping. Occasionally plants are found which have not been cleaned out since erection, as likewise have water tube boilers been found with scale thick enough in the tubes to stop the passage of water. But the point is that for efficient operation, and for the maintenance of capacity, the heat transfer surfaces must be in good shape.

Some engineers believe that after the first few months little or no oil should be added to the system, because, being a closed system, oil will travel around the cycle with the ammonia. This probably will work out all right with careful attention to details by experienced engineers—especially with the slower types of compressors. The oil trap should be kept drained of oil at all times, and the condenser and the expansion system should have a regular overhauling. However. as the oil which gets into the expansion pipes is likely to congeal, it is essential that proper means be used to get the oil to flow.

After *pumping a vacuum* on the cooling coils in order to free them of ammonia preliminary to overhauling (heating them if necessary) the coils should be drained of any remaining liquid and finally blown out with steam under pressure, giving a rather free exhaust so that the velocity of the steam will carry out the liquid and other soft deposit. After this steaming has been completed satisfactorily, hot air should be pumped into the pipes in order to dry out any moisture

remaining. Finally, in pumping a vacuum on the coils preliminary to recharging the system, the compressor should be run *slowly*. This is because with oil and air in the compressor there is needed only the proper mixture and the flash point temperature to secure an explosion. This flash point temperature is easily reached with vacuum pumps when one remembers the very high *ratio* of pressures possible under the circumstances.

Tests for Leaks. In order to keep the plant in good condition regular tests for leaks must be made. Ammonia test paper (sulfur paper) should always be on hand and used, and Nessler's solution should be available for periodic tests of the jacket water, condenser water and the brine. The condenser can be leaking with a split pipe for weeks without giving visible signs of trouble. The operator should approach the matter with an open mind, and remember that conditions are different in certain respects from those in the steam plant. In the case of steam the loss of 100 lbs. through the stuffing boxes, flanges, etc., probably would be of slight moment. With ammonia it would cause the loss of about $30 and probably would be objectionable on account of the odor. Just as in the case of the boiler plant—where continued care is necessary to prevent blow off valve leaks, leaky settings and air holes in the fuel bed—it is necessary always to be on the lookout to keep the refrigerating plant in good condition.

Suction Pressures. Unless the surrounding medium (air, brine, water, etc.) is hot enough the *ammonia will not boil.* The ammonia boiling temperature is a variable, depending on the pressure to which it is exposed, just as in the case of the steam in the boiler or the steam in the steam pipe or the radiator. If a brine tank has brine at 14 deg. F. it will not be possible to boil ammonia at a pressure of 30 lbs. gage because the boiling temperature of the ammonia at that pressure is slightly greater than 16 deg. F. Where only one expansion valve is in use the lack of any refrigerating duty would be in

evidence at once, but most plants have several loads with a variety of conditions, including cold storage, ice making and possibly a sharp freezer of one sort or another. In these

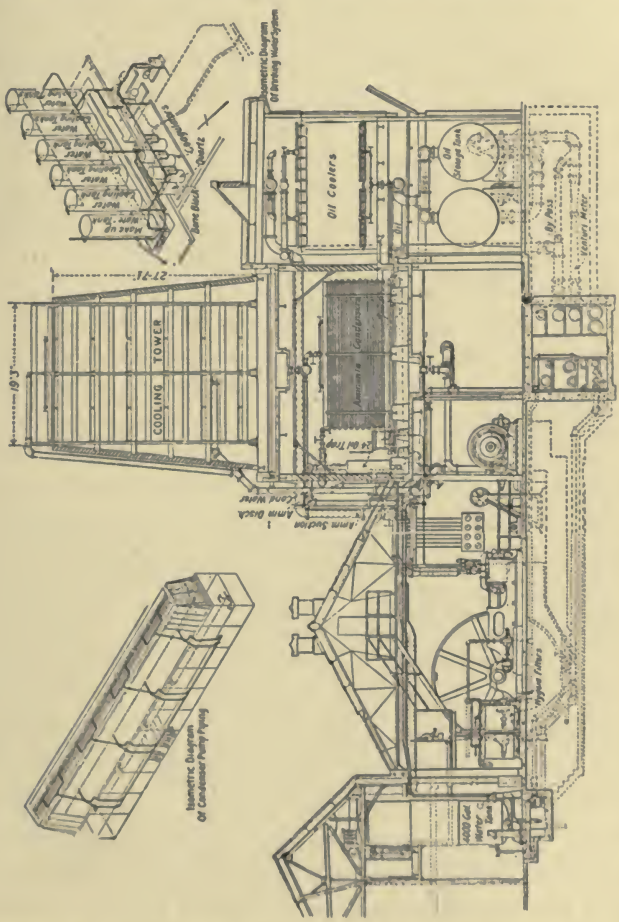


Fig. 65. Typical arrangement of a refrigerating plant showing the cooling towers and atmospheric condensers

plants, if the suction pressure is not low enough to boil the ammonia in the coldest room or tank, then the liquid feed to the coldest coils will not boil, but will accumulate and finally

the pipes will become flooded. "Lost" or "dead" ammonia is the result, combined with unsatisfactory operation. A condition such as has been mentioned is often found in the case of plants requiring a heavy load of short duration, like the chill of the packing plant or the dairy with its pasteurizing load, or even the plant which has to pull down the brine tank temperature in the morning. Incidentally it is wise to remember that at times it is possible to have a double suction, or to have one machine on the low temperature and the other machines on the medium temperature refrigeration. It is unwise to have to govern the conditions of the whole plant by the conditions prevailing in one room.

Defrosting and Lack of Capacity. As far as the routine work is concerned considerable advantage is derived from having a connection from the compressor direct to the expansion coils, or to the different sections of coils. To defrost or to start the oil flowing the hot gas from the compressor is allowed to flow directly into the low pressure side. The refrigerating coils then become a condenser for the time being without water cooling, and therefore they will rise in temperature. In consequence the solidified oil will become liquid and if the pipe line has no pockets, but drains towards its liquid trap the oil may be separated easily. Care must always be taken, however, to prevent slugs returning to the compressor. The same method of pumping the hot gas into the expansion piping may be used for defrosting. Sometimes oil in the lower (and colder) part of the condenser may be loosened by allowing the cooling water to be shut off for a short space of time. As an example of the advantage of overhauling the following typical example may be given:

Removing long accumulation of ice, the suction rose from 3 to 15 lbs. gage.

Removing long accumulation of oil in pipes, the suction rose from 15 to 17 lbs. gage.

By increasing the amount of piping, the suction rose from 17 to 22 lbs. gage.

The piping noted in the above was placed in some of the rooms which were under-piped. It is usually conceded that piping is the cheapest thing around the plant. The piping ratios should be correct for the kind of work being performed. As an example of piping and plant design, Figs. 65 and 66 are included.

To summarize, *inability to obtain capacity* may be brought about by:—

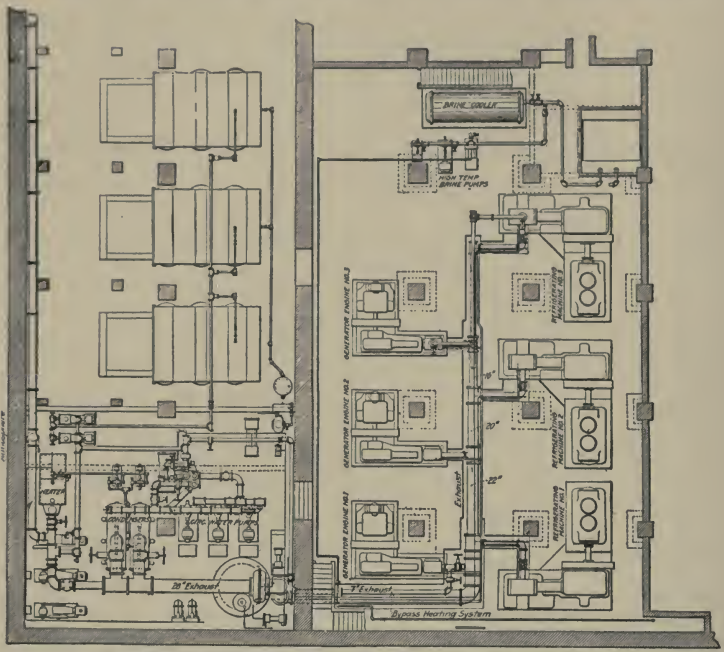
1. Too small a piping ratio, requiring a low suction pressure.
2. Frost on the outside, oil, scale, etc., on the inside of the expansion pipe.
3. Water in the expansion pipe, requiring a lower suction pressure for the required boiling temperature of the ammonia.
4. Leaky valves, and piston rings, that allow loss of *weight* of gas pumped by the compressor into the condenser.
5. Too small a charge, resulting in the blowing of gas through the expansion valve into the coils.
6. Suction pressure too low for the required refrigerating work being performed, or the suction valve partly closed.
7. Liquid returning to the compressor (wet compression).
8. Leaky safety head, or leaky or partly open by-pass valve.

Excess condenser pressure may be from:—

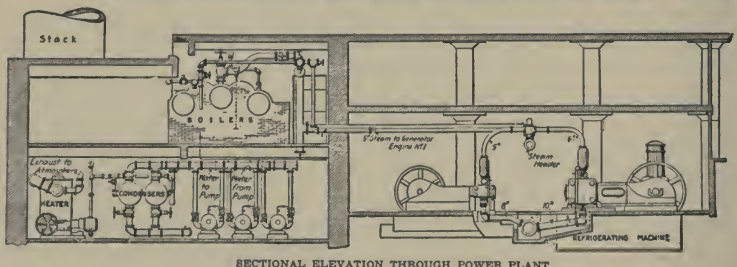
1. Too small a condenser, or one that is not working properly.
2. Too small an amount of water on the condenser for the temperature of water used.
3. Oil or scale on the condenser piping.
4. Condenser partly filled with liquid ammonia.
5. Stands of the condenser not working, due to valve trouble or accumulation of solids which prevent the condenser from draining.
6. Inert gases resulting from poor lubricant in the compressor, or air in the condenser due to pumping a vacuum during operation or to repair work.

Accidents and Their Prevention. It is also worth while mentioning a few of the most common causes of accidents. The rule in regard to steam boilers and steam pipe lines applies with even greater force to ammonia,—*don't break a joint under pressure.* Also the fact that a vacuum has been pumped on a set of coils is no reason for the coil to be free of ammonia. Install safety valves according to the A. S. R. E. Code (Fig. 67). Apply the "stick" test or use a blow torch on the pipe in order to boil away the liquid which may still be present. Do not pump an air pressure on the system in

too great a hurry, operate slowly and stop the machine occasionally. It is very easy to have too great a temperature of



GENERAL PLAN OF STEAM PIPING IN ENGINE, COMPRESSOR AND BOILER ROOM



SECTIONAL ELEVATION THROUGH POWER PLANT

Fig. 66. The boiler and engine room of a typical modern steam driven, medium speed, refrigerating plant

discharge, which may be the only other prerequisite for an explosion. In large plants install automatic stop valve (Fig.

68), which should be installed in the main pipe lines. In breaking a joint be careful to do it slowly. It is better to loosen and still have bolts in the flange until a lack of pressure in the pipe is found actually to exist. Don't fill a drum or receiver or system of piping full of cold ammonia and prevent or try to prevent expansion by closing all valves. The container will burst every time when the temperature rises. Don't operate at a pressure below the atmosphere—

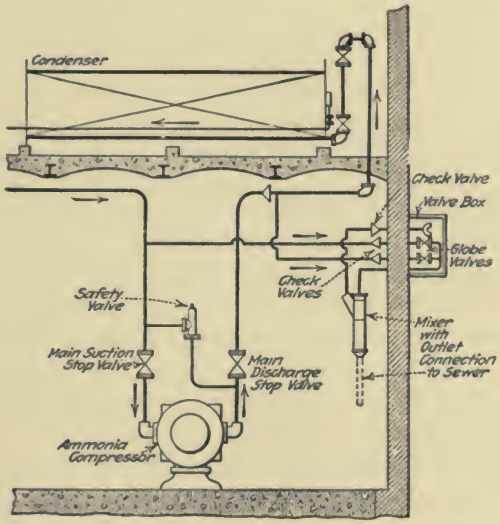


Fig. 67. The A. S. R. E. excess pressure safety valve piping arrangement and mixer

unless absolutely required—and don't operate the machine while the pipe line is open for repairs, and make sure that the joint is tight before starting up again. Ammonia does not usually *disintegrate*, and does not explode except with particular mixtures of air, but on the other hand, it is useless to run any unnecessary danger.

Starting Up the Compressor. In starting up the *compressor* it is usual to proceed as follows:—

1. Start the condensing water over the condenser, and through the compressor jacket.
2. See that both the suction and the discharge valves on the compressor are closed, *open* the discharge by-pass to the suction line.
3. See that all lubricators are open and that the oil supply is ample.
4. Start the compressor—slowly—if it is possible so to do.
5. Open the suction and the discharge valves.
6. Close the by-pass valve.

Operate so as to pull down the suction pressure to the amount desired, then open the expansion valve. Try to bring the frost back to the suction bends, but do not frost the machine. If the machine is motor-driven remember that the power required to operate at high back pressure is greater than it is at low back pressure (this is the opposite from what might be expected). For example, at 170 lb. gage pressure the *mean effective pressures* are as follows:—

At 5 lbs. gage suction pressure the mean effective pressure is 58.0 lbs.
At 10 lbs. gage suction pressure the mean effective pressure is 63.5 lbs.
At 15 lbs. gage suction pressure the mean effective pressure is 67.7 lbs.
At 20 lbs. gage suction pressure the mean effective pressure is 70.9 lbs.
At 25 lbs. gage suction pressure the mean effective pressure is 73.3 lbs.
At 30 lbs. gage suction pressure the mean effective pressure is 75.0 lbs.

Sometimes the suction pressure is high at the start, and the power required by the compressor is 40 per cent greater than under standard conditions. To shut down the machine reverse this order of operation.

Starting and Operating the Absorption Machine. In operating the *absorption machine* the cycle must be fully understood if one is to get results of value (see Chapter IV). Briefly the proper steam pressure must be used in the generator for the average concentration of aqua carried. The strong aqua pumped back by the aqua pump must be 6 to 12 per cent or more higher in concentration than the weak aqua leaving the generator, and the pump must be operated so as to provide the necessary amount of aqua for the concentrations being used. To operate (assuming there is pressure on the generator), proceed as follows:

1. Turn on the water supply to the condenser, absorber, rectifier, and the weak aqua cooler.

2. Start the strong aqua pump, and see that the weak aqua is flowing properly, and the weak aqua regulator on the strong aqua receiver is working.
3. Start the brine pump, if one is installed. Note the ammonia pressure in the brine cooler (for expansion coils) and in the absorber. Remember that in this machine the boiling temperature of the ammonia is determined by the pressure in the absorber.
4. Turn the steam on the generator, using the steam pressure required by the consideration of the concentrations carried in the generator. When the supply of anhydrous ammonia is sufficient in the receiver, and the absorber pressure is correct for the load being carried, then:
5. Open and adjust the expansion valve.

In shutting down reverse the order of operations.

In *operating* the absorption machine the main difficulties consist in keeping up the concentration of the aqua in the generator, and in keeping water out of the expansion coils. It should be remembered that, as in the compression system some oil gets into the liquid receiver and then into the expansion coils, that the rectifier cannot remove all of the water vapor out of the gas boiled out of the generator, and that in time the coils receive more water and the result is that these become partly filled with a *strong aqua.* The result is that the coils must be pumped out, and the operation must be attended to frequently enough to keep the low pressure part of the system in good condition. Also as the ammonia is lost through leaks or disintegration or otherwise the concentration must be brought up to normal by means of a new charge or the steam pressure must be raised. More inert gases are formed in the absorption machine than in the compression system, and purging must be resorted to to keep the condenser pressure down to normal. It is now recommended that a solution of potassium bichromate using 1/5 of a pound per 100 lbs. of aqua be used in the generator to reduce the action of CO_2 and the formation of oxide of iron sludge and of free hydrogen in the generator.

After testing the new pipe line with air pressure, and soap suds, without evident leaks the air pressure should be maintained for 24 hours without appreciable drop. The system is then ready for charging, and the first operation is to free

the pipe line of air—usually performed with a free exhaust to the atmosphere so arranged as best to free the line of dirt and accumulated impurities. To pump a vacuum the com-

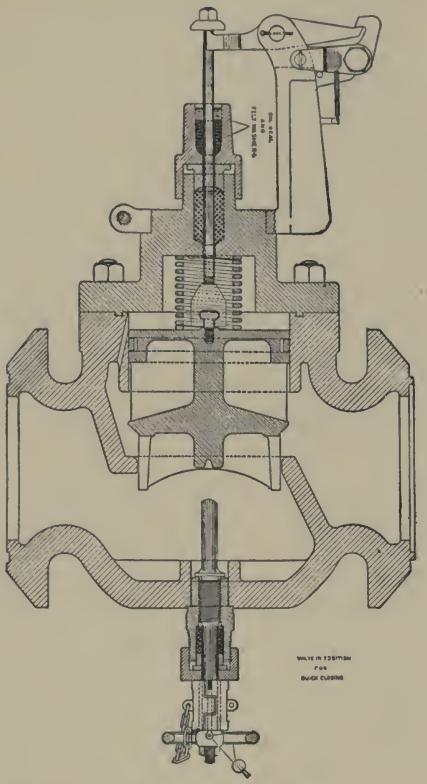


Fig. 68. Semi-automatic stop valve for ammonia lines. [Note lever and catch at top]

pressor exhaust valve is closed, the compressor is operated, and the exhaust is allowed to escape into the atmosphere through some suitable opening between the compressor and the exhaust stop valve. The compressor is operated slowly

as a vacuum air pump until a vacuum within 4 inches of the barometer is obtained. The system is now ready for charging.

Charging With Ammonia. The ammonia charge is made by connecting the drum of ammonia to a point in the pipe line between the liquid receiver and the expansion valve where the charging connection has been placed. In charging the drum is placed on a platform scales and the amount of ammonia drawn out of the drum may be known by referring to the weights on the shipping tag and the reading on the scales. The liquid should be allowed to flow into the expansion system only (not into the liquid receiver) and the valve on the drum becomes the expansion valve during charging. The ammonia will flow readily because the pressure in the drum is that corresponding to the pressure of saturation (practically) given in the ammonia tables for the temperature in the room, and of course the suction pressure is readily controlled at about 15 to 20 lbs. by the operation of the compressor. Continue the action of feeding into the system and operating the compressor until a pressure in the condenser is obtained approximately as follows:

For 60 deg. F. water on the condenser the condenser pressure should be 150 gage.
For 65 deg. F. water on the condenser the condenser pressure should be 160 gage.
For 70 deg. F. water on the condenser the condenser pressure should be 170 gage.
For 75 deg. F. water on the condenser the condenser pressure should be 180 gage.
For 80 deg. F. water on the condenser the condenser pressure should be 190 gage.

And of course the actual amount of ammonia charge will depend on the amount of piping. This is approximately ⅛ lb. per lineal foot of 1¼ inch pipe, 1/5 lb. per lineal foot of 2 inch pipe when the pipe is ordinary expansion, and ¼ lb. and 0.7 lb. respectively when the pipe is flooded. For condensers use 15 lbs. per coil (12 pipes high) of 1¼ x 2 inch double pipe, and 30 lbs. per coil (12 pipes high) for 2 inch atmospheric condensers. We usually figure to fill the re-

ceiver half full. The charging drum is empty when its weight is that given as the "tare" on the shipping tag. Such a number of drums must be used as will give a sufficient charge of ammonia. The refrigerating plant is now considered to be ready to operate.

Cooling Water

Refrigeration and *cooling water* are inter-related terms, for without cooling water there could not be refrigeration. One would be as likely to operate a steam engine or steam turbine condenser without condensing water, or the oil engine without cooling water, as the refrigerating machine without the necessary amount of water at the temperature of the air or colder. As a matter of fact, the first question that arises when considering the installation of a new plant is, "What is the temperature of the cooling water?" and the second is, "What is its cost?" In other words, the engineer desires to find out how fully it is possible to use the cooling water and what will be the cost of providing the required amount.

Occasion has been taken before this to bring out the *function* of the cooling water in the refrigerating cycle. The compressor is a pump—a temperature pump in a way—as it raises the temperature level of the refrigerant to a point sufficiently high so that the cooling water at hand can condense the ammonia gas satisfactorily. In the ammonia condenser, as in any form of condenser, the cooling water removes the heat of liquefaction (latent heat) and in addition the superheat and some of the heat of the liquid. In consequence of this the water is warmed a certain amount, depending on the relative amounts of water used. In general it may be said that the heat removed is as follows:

The compressed ammonia gas, being superheated, has its superheat removed first and then its latent heat of liquefaction. Finally, the liquid ammonia is precooled slightly below the temperature of saturation. The cooling water has to

remove all of this heat, and then to pass away from the condensers carrying this amount of heat with it at a proportionately higher temperature. An example will make this clearer:

Example. Take a compressor receiving ammonia gas in a dry, saturated condition and operating between pressures of 15.7 lbs. gage and 175 lbs. gage, with a temperature of discharge of 234 deg. F. and a temperature of the liquid leaving the condenser of 80 deg. F. The temperature of the water leaving the condenser is 90 deg. and the water entering is 75 deg. F. Under these conditions there would be removed for each pound of ammonia 92 B.t.u. of superheat and 485.5 of cooling would be needed to condense it at 175 lbs. gage pressure and a temperature of saturation 93.1 deg. F. Finally to cool the liquid to 80 deg. F. requires 15.1 B.t.u. or a total of 592.6 B.t.u. per pound of ammonia. The useful available refrigeration from this same pound of ammonia is 479.6 B.t.u. and the difference, 592.6—479.6=113.0 B.t.u. is the work done by the compressor per minute, resulting in heating the ammonia. This result may be shown in another way.

Taking a case where a pound of ammonia is compressed per minute, the net refrigerating effect is 479.6 B.t.u. or a rate of 2.40 tons of refrigeration. With these conditions of operation there will be required (with no allowance for losses) 1.125 h.p. per ton of refrigeration or

$$1.125 \times 2.40 \times 42.4 = 114.5 \text{ B.t.u.}$$

per pound of ammonia condensed, in the condenser. The difference of 1.3 per cent in the two calculations is due to errors in the constants used in the calculations, but the example shows the real function of the condenser and of the cooling water. It is frequently said that the condenser must remove an amount of heat equivalent to the amount of useful refrigeration done *plus* the heat equivalent of the work done by the compressor. This likewise may be seen by the example as noted above. The total heat removed by the condenser

per pound of ammonia was 650.9 B.t.u. and the tonnage under the conditions of the problem was 2.40 and the quotient, $\frac{592.6,}{2.40}$ is 247 B.t.u. per ton per minute. If a 100 ton plant is being figured on them 24,700 B.t.u. must be removed per minute, which (at 1¾ gallons of water per ton per minute) will require 175 gallons per minute of cooling with a temperature rise of about 17.0 deg. F.

To supply the required amount of condensing water is frequently a difficult and expensive job, unless the plant is

Fig. 69. Hadley Ice Co., Chicago, Ill.

situated near a plentiful supply of satisfactory water. It frequently occurs that the plant is located in such a way as to limit the water supply to wells or to the city supply. The cost of water from the city mains is usually beyond the economical means of the plant, and wells may be expensive to dig. For example, some of the wells in the stock yards district of Chicago were sunk to 1,600 feet, and cost $14,000. So, if water is expensive to pump, and has to be purchased, a method of cooling would be necessary to make it serve over

again one or more times in the condenser. The methods used are as follows:

In general there are three ways of cooling water where reuse of such water is desired for cooling purposes; by the use of *cooling ponds, spray ponds* and *cooling towers.* Each of these methods has its own advantages in special cases.

The Cooling Pond. The cooling pond can be used in locations where a natural body of water is available adjoining the plant. The water is cooled by being allowed to be brought into surface contact with the air, and the evaporation (and the cooling of the water) depends on the relative temperature of the air and on the humidity. The tendency would be for the pond to assume the temperature of the air both due to giving up heat to the air and also due to the difference of the *saturation pressure* of the water vapor in the air. Roughly the amount of cooling surface for usual conditions may be taken at from 60 to 80 B.t.u. per square foot of surface per hour, or about 225 sq. ft. of cooling surface per ton of refrigeration. As the surface is the main consideration, it is not important to have any depth unless the refrigerating load is a variable amount, as, say, in the cases already mentioned of the chill rooms of packing houses, or in the case of pasteurizing milk or cream. In these latter cases it would be possible to use a smaller pond, but of some depth in order to store up enough cooling water. Usually a foot or two is plenty.

Spray Ponds. Spray ponds differ from cooling ponds in the attempt that is made to accelerate the cooling effect. By forming a spray and by very finely dividing the particles of water, the air may quickly reach all the water instead of waiting until each particle reaches the surface of the cooling pond—a process that might be a lengthy one under adverse conditions.

With the *spray nozzle* (Figs. 69 and 70) the water is pumped under 5 or 10 lbs. pressure through a special nozzle

designed to create a whirling spray of fine water particles. In most cases a cooling of 10 to 15 deg. F. may be obtained with a single spraying, and the final temperature depends on the fineness of the spray and on the humidity. For instance, if the air has a humidity of 70 per cent and a temperature of 70 deg. F. the pressure of the water vapor in the air is 0.70 times 0.36 (the saturation pressure of steam at 70 deg. F.) or a pressure of 0.25 lb. per sq. in. It is usually assumed that the cooling water may be reduced to a

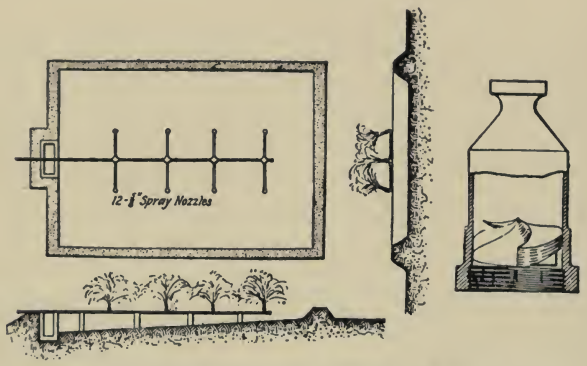


Fig. 70. Arrangement of sprays for a spray pond. Note size of basin required to prevent loss of water by drift. [15 ft. is required from nozzle to edge of pond]

temperature corresponding to a pressure of about 0.15 lb. sq. in. greater than that of the water vapor in the air, or (in this case) $0.25 + 0.15 = 0.40$ lb., which corresponds to the temperature of 73 deg. F. Some five or six per cent of the water sprayed is evaporated in the process.

A good rule to use is to allow one square foot of surface for every 250 to 300 lbs. of water sprayed per hour, which reduces to about 4 sq. ft. of surface per ton of refrigeration, allowing a temperature range of the water of 12 deg. F. To prevent excessive loss of water due to the drift of the spray

during moderate winds the pond or catch basin should have the embankment at least 15 ft. outside the row of spray nozzles. The nozzles should be from 8 to 10 ft. apart to secure satisfactory results. The spray pond may be placed on the roof or above the condenser if suitable louvres are provided to prevent excessive loss of water during winds. Fig. 71 shows a good arrangement for gravity feed from the spray pond to the troughs over the condensers.

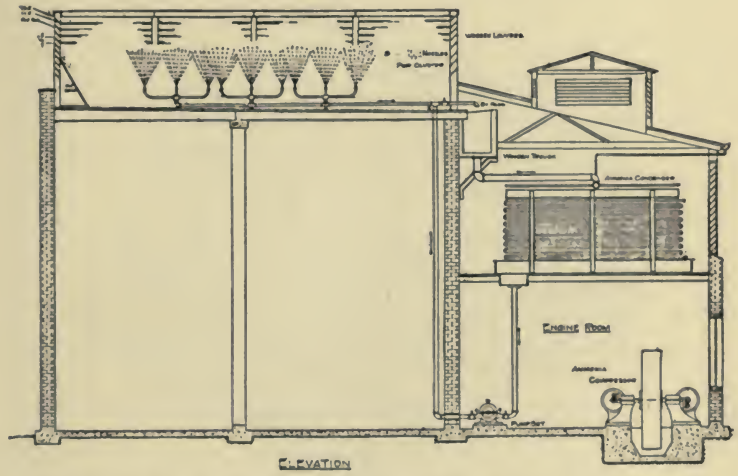


Fig. 71. A typical spray arrangement for the roof. Note louvres, catch basin and gravity feed to the atmospheric condenser, and position of centrifugal pump

The Cooling Tower. The cooling tower has been successful and popular for many years, and without question will continue to be used indefinitely with equal satisfaction in spite of the success of the spray pond in particular instances. It lends itself particularly to congested quarters, where the cooling pond is out of the question and the spray nozzle cannot be easily arranged. As the name would imply, it is built in the form of a tower usually with air tight walls to gain

the advantages of the chimney construction. The object of the tower design is to acquire a natural or a forced draft (or both) and to provide a means of securing a large amount of wetted surface kept wet by the condensing water that is pumped in at the top.

The *wetted surface* may be secured by means of tile, but the usual manner in refrigeration is to form a wooden checkerwork, arranged as in Fig. 72, to prevent undue trouble

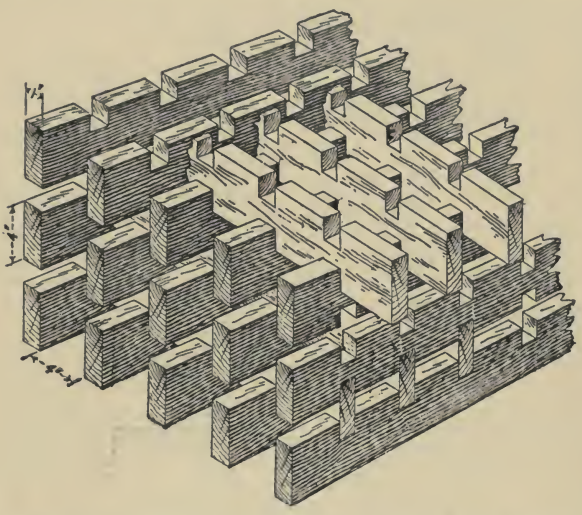


Fig. 72. One method of building wooden cooling towers. [The idea is to secure as efficient surface as possible]

by warping and so that the wet cross sectional area will be sufficient to allow the necessary amount of air to pass through. As in the spray nozzle, which requires a finely atomized spray, so in the cooling tower a thin film of water is required and an air current to evaporate some moisture and to cool the water by the heating of the air.

The *action* of the cooling tower is as follows: The water is showered over the top grids of the tower, and it descends

dropping from checkerwork grid to checkerwork grid and being exposed all the time as a thin film on the surface of the tower. The ascending air absorbs moisture, has its temperature increased, and finally leaves the tower at a temperature of about 10 deg. F. below the temperature of the entering warm water from the ammonia condensers, and with a humidity at the exit temperature of about 95 per cent. During the passage of the air through the tower some water is evaporated, which cools the remaining water according to the principles of refrigeration.

In refrigeration it will be necessary frequently to cool water about 15 to 20 deg. F. Under usual conditions the tower will absorb heat from the water at the rate of 300 to 400 B.t.u. per square foot of tower wetted surface per hour. At this rate 40 to 50 sq. ft. is required per ton of refrigeration and an air velocity in the tower of about 10 ft. per second is expected. In refrigeration a natural draft tower is generally used, and either the necessary chimney effect is relied on for the creation of the draft, or the design with open sides is used, which relies on the wind to do the business.

It will thus be seen what the function of the cooling water is and what means of making a repeated use of this cooling water is possible. As a means of visualizing the problem better and also of estimating the requirements of new plants the diagram, Fig. 73, is added showing the heat to be removed by the condenser per ton of refrigeration, and the condenser pressure to be expected in this condition for each temperature range of the cooling water.

Example. For example if 2.45 gallons of water are used per ton of refrigeration per minute with a suction pressure of 15 lb. gage, the range of the water (the temperature rise of the water in passing through the condenser) will be 12 deg. F., the heat removed per ton per minute will be 244.9 B.t.u. and the condenser pressure will be 158 lb. gage. The diagram is laid out for an initial temperature of 70 deg. F. and no

liquid cooling is allowed for after condensation of the ammonia. Also the assumption is made that the gas entering the compressor is just dry and saturated (a condition that prevails only with careful operation of the compressor by means of thermometers) in which state the maximum efficiency of the compressor is also obtained. The assumptions enumerated are average figures, and the chart becomes much simpler than it would have been in order to include all variables.

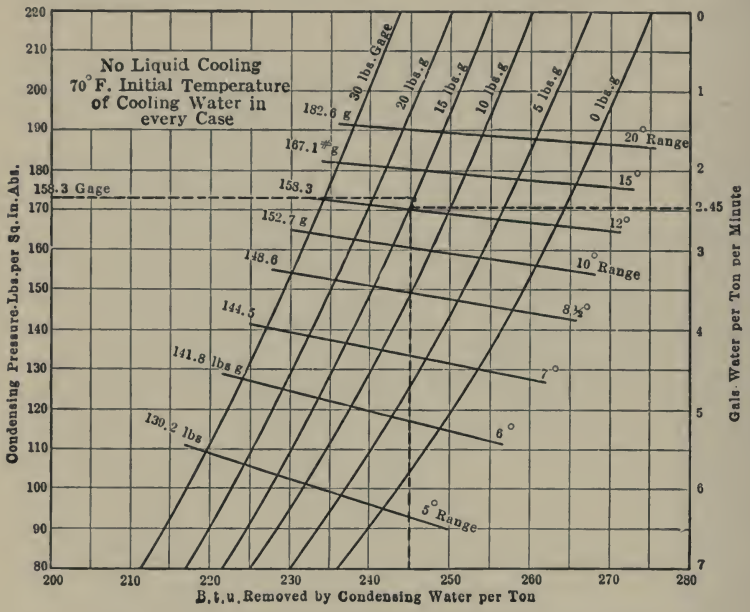


Fig. 73. Chart for calculating condenser performance

The chart shows in an excellent manner the effect of changing the conditions of operation. It is evident that some advantage is derived by the use of more water. For example, if 5.8 gallons of water were used with 15 lbs. suction pressure the condenser pressure would be 139.2 lbs. per sq. in. gage with a temperature rise of 5 deg. F. and a removal by the

water of about 228.5 B.t.u. (see intersection of the 139.2 condenser pressure and the 15 lbs. suction line) per ton of refrigeration per minute. The reduced amount of heat removal is, of course, on account of *less work* done by the compressor as the pressure is reduced from 158 lbs. to 139.2 lbs. The addition, then, of over double the amount of water (from 2.45 gallons with 12 degree range to 5.8 gallons with a five degree range) had the effect of reducing the head pressure only 18.8 lbs. and the work done in the compressor by only about 16.4 B.t.u. Since the advantage derived by showering the condenser with water is not very great apparently, its value may be questioned.

Conclusion. In this chapter the principal causes of grief and trouble have been given, with also a description of the factors influencing economy of condensing water. Erection must be more carefully attended to than is usual in steam engineering, and in operation the engineer has to maintain his system tight as well as prevent accumulation of foreign material tending to prevent the proper functioning of either side of the system. The refrigerating machine may be excellent, and the piping adequate and yet the plant may be a failure unless details of operation are strictly attended to.

CHAPTER VI

Refrigerants

It is quite possible that there has been some concern on the part of engineers as to why ammonia machines are used for refrigeration in the United States to the extent of more than 95 per cent of all such work, whereas in Great Britain and France other cooling vapors are used to a much greater extent. There must be some reason for this preference—some good and sufficient points of excellence—for engineers in the United States have always been very quick to make changes if there is anything to be gained thereby. What are the reasons for the preference for ammonia, and why do not carbon dioxide (CO_2), sulphur dioxide (SO_2), ethyl chloride and other vapors gain in popularity?

Analysis of Choice. All *engineering questions* have two sides, and few if any live matters can be answered definitely and decidedly one way or the other. As an example, there is the question of the use of the jet or the surface condenser for stationary plants (there is no question as to the use of the surface condenser in marine practice). There is the question of the use of star or delta windings in electric machinery or of two phase or three phase electrical transmission, and the question of the steam engine or the steam turbine as compared with the Diesel oil engine, and both of these as compared with the development of water power—if the latter is available. The questions of load factor, cost of fuel, cost of money (if borrowed) kind and quantity of

water, proximity and quality of fuel to be used, etc., are factors that must be weighed in the balance if a satisfactory answer is to be obtained. No cut and dried answer will avail in all cases, but each individual case must be decided for itself.

The refrigerating engineer understands that our *objective* with the use of the compressor and the high pressure side is to be able to use the same refrigerant continuously. We compress the gas boiling off from the refrigerating coils in order that the condensation of the gas may be possible at temperatures of 60, 70 or 80 deg. F. or even higher, depending on what temperatures of cooling or condenser water may be obtained. The gas compressed passes through the compressor, and the amount of gas so compressed per interval of time depends on the piston displacement of the compressor. Naturally we are interested not only in the pressure range of the compressor (the suction and discharge pressures required by the refrigerant used), but also in the cooling effect obtainable by one cubic foot of compressor displacement.

Chemical Properties of Refrigerants. There are in addition the factors of the cost of the substance used, the pathological effect should the stuff escape from the compressor or cold storage piping and the chemical effect on iron, steel and brass. Of these the first item, the cost of the refrigerant, is too great in every case to allow the substance to be thrown away. If air were used it might be possible to exhaust it into the atmosphere after using, but air is no longer employed. The effect on human life of the gas used is different with each substance. Carbon dioxide is harmless in small quantities and is odorless when pure. In large quantities its effect on one is practically the same as if one were submerged in water. Sulphur dioxide and ammonia are deadly even in relatively small quantities if relief is not immediately obtained. Ethyl chloride in small quantities seems to be the least harmful of all refrigerants now in use, and for that

reason it is used to a large extent on naval vessels for medium refrigerating temperatures. The result in all cases is that careful provision has to be made for preventing leaks or breaks in the pipes or headers. Also it is necessary to provide for the withdrawing of the charge into drums or for

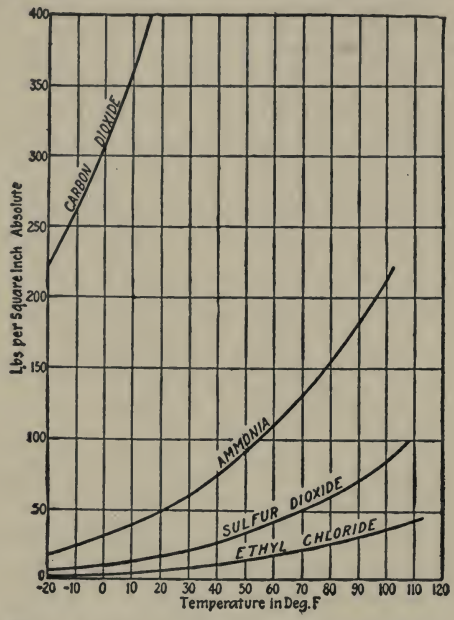


Fig. 74. The presure-temperature relations for the common refrigerants. [Note the relative pressure at, for example, 80 deg. F.]

storage in another part of the system when repairs or cleaning of any portion of the plant have to be performed.

The Physical Properties. The main difference in refrigerants, and (with the exception of ethyl chloride for use on shipboard) the deciding factor in its choice, lies in the *physical properties* as regards the pressures usually obtained during operation, the refrigerating effect of one pound of

the substance and finally the volume of one pound of the refrigerant at the pressure of boiling in the expansion coils.

Referring to Fig. 74, it will be seen that ethyl chloride has the least and sulphur dioxide has very moderate pressures corresponding to the temperatures of the cooling coils or the condenser, but that carbon dioxide has (relatively) extremely heavy pressures—some 200 to 300 lbs. for suction and 800 to 1,000 lbs. per sq. in. for the condenser pressure. These latter pressures are so great as to require special materials both for the compressor and the piping and this

TABLE 7

	Refrigeration [Net] Per Lb. at 0° F. Boiling Temperature	Specific Volume of Dry, Saturated Vapor at 0° F.	Theoretical Volume of Piston Displacement for 100,000 B.t.u. Refrigeration	Column Three Taking CO_2 as Unity	Suction Pressure Lbs. Per Sq.In. Gage	Condenser Pressure Lbs. Per Sq.In Gage
Ethyl chloride	160.6	19.0	11900	32.50	—10	8.2
Sulphur dioxide	149.34	7.49	5020	13.72	— 4.35	45.
Ammonia ...	491.1	9.114	1855	5.07	15.7	126
Carbon dioxide	77.0	0.282	366.0	1	288	891

Refrigeration is at 0° F.
Temperature at expansion valve 70° F.

factor is one that has been a deciding point in all save a few particular cases in the United States. In fact, unless great care is exercised the loss of the refrigerant becomes excessive, frequently to an extent such as will make the replacement cost greater than for ammonia. As will be seen, the small size of the compressor using carbon dioxide and the fact that the gas is not necessarily dangerous immediately, if it escapes, has caused it to be used in many places where space is valuable and where congestion makes it doubly dangerous to use ammonia. Fig. 74 also shows that ammonia

has moderate pressures, both at zero degrees F. (for the cooling coils) and at 70 to 80 deg. F. (for the condensing piping). The pressures encountered are nominal and are those which steam engineers are used to. The substance, however, has greater ability for escaping out of confinement than has steam or air, and also, it has a corroding action on copper and brass, and therefore the materials used are special air furnace cast iron of extra thickness, steel forged fittings, or high grade semi-steel castings, and either extra heavy steel or wrought iron pipe. The joints have to be special, either screwed fittings using litharge and glycerine or solder joints or flanged joints using tongue and groove with lead or rubber gaskets.

Piston Displacement. However in the final analysis the real points to be considered are the *horsepower input* per ton of refrigeration and the cubic feet of piston displacement of compressor per ton. All other factors fade in comparison unless the conditions are very extreme. Of course it is desired to produce refrigeration at as small cost of power as practical, and this cost of power is mostly in the demands of the compressor, which *lifts* the gas from the low side to the condenser. Again, as a rule, the volume of gas handled affects not only the size and cost of the compressor, but of the piping as well.

Referring to Table 7 it will be seen that the *volume displaced* by the compressor is less for carbon dioxide than for any other of the refrigerants. In fact, ammonia is 5.1, sulphur dioxide 13.7 and ethyl chloride is 32.5 times as great as carbon dioxide for similar conditions. For the same piston speeds the areas of the respective cylinders would have to have these same values, and the diameters would have to be 2.22, 3.7 and 5.71 times as great as for carbon dioxide. Ethyl chloride is fortunate on account of the small number of operating troubles. The low side, being subject only to an inward pressure, will leak air into the system if leaks should occur but leaks are readily prevented with proper

designs and careful operation. When air does accumulate, it collects in the condenser, from which it can be purged in the usual manner. The gas, however, is not entirely harmless and, if taken in large enough quantities, will put one to sleep like any anesthetic. Mixed with about twice its volume of air it forms an explosive mixture. The vapor burns readily with a greenish-edged flame. On account of the extremely large volumes at usual refrigerating temperatures, its application has to be limited to special cases and at present to

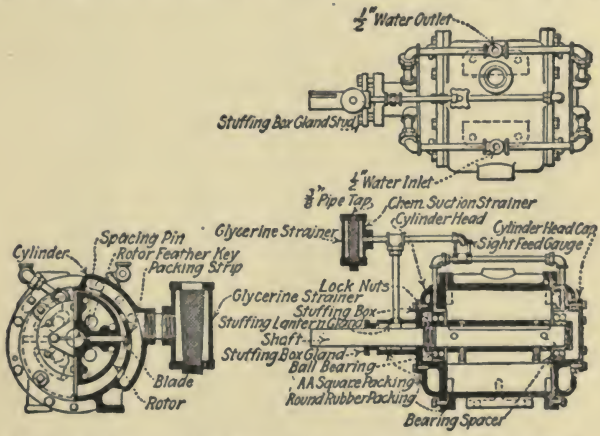


Fig. 75. The rotary form of compressor, used for ethyl chloride. This design has been successful in the U. S. Navy

rather small machines. The pressure on the high side being very small it lends itself easily to the use of the blower or the centrifugal compressor. The centrifugal compressor (for ethyl chloride, Fig. 75, and other refrigerants) certainly is going to be used to a much greater extent in the future, especially as direct connection to steam turbines and the cheaper (higher speeds) electric motors is possible with reduced losses.

Sulphur Dioxide. Sulphur dioxide has nominal pressures and like carbon dioxide does not have a bad effect on copper

and brass. Sulphur dioxide is to some extent self-lubricating, whereas ethyl chloride requires glycerine. The great objec-

TABLE 8. SATURATED AMMONIA: TEMPERATURE TABLE
[Bureau of Standards]

Temp. F.	Pres. Lb. Per Sq. In.	Spec. Vol. Cu. Ft. Per Lb. Liquid	Vapor	Total Heat of Liquid	Latent Heat of Evap.	Total Heat of Vapor
—40	10.41	0.02322	24.84	0.0	597.0	597.0
—35	12.05	0.02333	21.66	5.3	593.6	598.9
—30	13.91	0.02345	18.96	10.7	590.2	600.9
—25	15.98	0.02357	16.64	16.0	586.8	602.8
—20	18.30	0.02369	14.67	21.4	583.3	604.7
—15	20.88	0.02381	12.97	26.7	579.7	606.4
—10	23.75	0.02393	11.50	32.1	576.1	608.2
— 5	26.92	0.02406	10.22	37.5	572.4	609.9
0	30.42	0.02419	9.114	42.9	568.7	611.6
1	31.16	0.02422	8.911	44.0	567.9	611.9
2	31.92	0.02424	8.712	45.1	567.2	612.3
3	32.69	0.02427	8.520	46.2	566.4	612.6
4	33.48	0.02430	8.332	47.2	565.7	612.9
5	34.28	0.02432	8.149	48.3	564.9	613.2
6	35.09	0.02435	7.971	49.4	564.1	613.5
7	35.92	0.02438	7.797	50.5	563.3	613.8
8	36.77	0.02440	7.628	51.6	562.6	614.2
9	37.63	0.02443	7.464	52.7	561.8	614.5
10	38.51	0.02446	7.304	53.8	561.0	614.8
11	39.40	0.02449	7.148	54.9	560.2	615.1
12	40.31	0.02451	6.996	56.0	559.5	615.5
13	41.24	0.02454	6.847	57.0	558.7	615.7
14	42.19	0.02457	6.703	58.1	557.9	616.0
15	43.15	0.02460	6.562	59.2	557.1	616.3
16	44.12	0.02462	6.425	60.3	556.3	616.6
17	45.12	0.02465	6.291	61.4	555.5	616.9
18	46.13	0.02468	6.161	62.5	554.7	617.2
19	47.16	0.02471	6.034	63.6	553.9	617.5
20	48.21	0.02474	5.910	64.7	553.1	617.8
21	49.28	0.02476	5.789	65.8	552.3	618.1
22	50.36	0.02479	5.672	66.9	551.5	618.4
23	51.47	0.02482	5.557	68.0	550.7	618.7
24	52.59	0.02485	5.445	69.1	549.9	619.0
25	53.73	0.02488	5.336	70.2	549.1	619.3
26	54.90	0.02491	5.230	71.3	548.2	619.5
27	56.08	0.02494	5.126	72.4	547.4	619.8
28	57.28	0.02497	5.024	73.5	546.6	620.1
29	58.50	0.02500	4.925	74.6	545.8	620.4
30	59.74	0.02503	4.827	75.7	544.9	620.6
35	66.26	0.02518	4.375	81.2	540.7	621.9
40	73.32	0.02533	3.974	86.8	536.5	623.3
45	80.95	0.02548	3.616	92.3	532.1	624.4
50	89.19	0.02564	3.296	97.9	527.7	625.6
55	98.05	0.02581	3.010	103.5	523.1	626.6
60	107.6	0.02597	2.754	109.2	518.5	627.7
65	117.8	0.02614	2.522	114.8	513.7	628.5
70	128.8	0.02632	2.314	120.5	508.9	629.4
75	140.5	0.02650	2.126	126.2	504.1	630.3
80	153.1	0.02668	1.956	132.0	499.1	631.1
85	166.4	0.02687	1.803	137.7	494.0	631.7
90	180.7	0.02707	1.663	143.5	488.8	632.3
95	195.8	0.02727	1.536	149.3	483.5	632.8
100	211.9	0.02747	1.419	155.2	478.0	633.2

tion to sulphur dioxide is the corroding action on iron and steel whenever moisture gets into the system. The com-

bination of water and sulphur dioxide forms sulphurous acid. The result is that the sulphur dioxide (SO_2) machine has to be very self-contained and sealed against air leakage. In this country SO_2 machines are used only in the smaller sizes and seldom, if ever, used where direct expansion is employed in the sense that is usually understood. To do so not only would require large return mains, but would greatly increase the possibilities of corrosion.

Carbon Dioxide Carbon dioxide (CO_2) has its particular field in congested localities, such as office buildings, hotels

TABLE 9. PROPERTIES OF SULPHUR DIOXIDE
Cailletet and Mathias

Temp. Deg. F.	Pressure Lbs. per Sq. in. Abs.	Specific Volume Cu. Ft.	B.t.u. Heat of Liquid Above 32° F.	B.t.u. Latent Heat of Vaporization	B.t.u. Total Heat
—20	5.92	12.98	—15.69	175.63	159.94
—10	7.90	9.51	—12.85	173.87	161.02
0	10.35	7.49	—9.91	171.83	161.92
5	11.80	6.66	—8.38	170.68	162.30
10	13.40	5.91	—6.84	169.43	162.59
15	15.15	5.21	—5.32	168.19	162.87
20	17.09	4.64	—3.78	166.90	163.12
25	19.17	4.14	—2.21	165.51	163.30
30	21.46	3.72	—0.62	164.05	163.43
40	26.82	3.01	+2.57	160.99	163.55
50	33.38	2.43	5.85	157.60	163.45
60	40.53	2.00	9.20	154.00	163.19
70	49.56	1.65	12.58	150.2	162.7
80	59.58	1.36	16.09	146.0	162.1
100	84.75	0.95	23.17	137.0	160.2

and marine installations where almost invariably the indirect system of brine circulation is used. This allows centralization of the high pressure piping, and the use of common black pipe for the refrigerating piping. An escape into the air of a charge of carbon dioxide is not so disagreeable or subject to such unfavorable notice as ammonia would be. As it is much heavier than air, it will not rise except by

diffusion, and guests or tenants are not driven out of their rooms. Also unlike ammonia or ethyl chloride it is not a combustible nor will it form explosive mixtures with air as the other two will do. On the other hand it is preferably a cold cooling water refrigerant. Its critical temperature is 88.4 deg. F., which means that in extreme cases in the semi-tropics or for work in certain locations which have very warm condensing water it is possible that the discharge gas passing into the condenser will not be condensed, but will act somewhat like the "dense air" refrigerating machine which was used in the government service for many years. Under such conditions the efficiency is reduced, and better results could have been obtained by the use of some other refrigerant.

It so happens that, contrary to the expectation of most engineers, the power required by the different refrigerating machines per unit of refrigeration is about the same. One might expect that carbon dioxide, working as it does with a pressure range of about 550 lbs. would require much more power input than would, say, sulphur dioxide, which works between the limits of -5 lbs. and 35 lbs. or only 40 lbs. total pressure. However, the volumes to be pumped are to be considered, and there are 13.7 times the volume of SO_2 to be pumped for the same refrigerating effect as for CO_2 at zero degrees F. The net result is that carbon dioxide requires a little more power than does ammonia or sulphur dioxide. In fact, ammonia leads slightly in this respect but the net result is so close that the matter of the horsepower per ton of refrigeration for usual (standard) operating conditions need not be considered when deciding on one or the other refrigerant.

To sum up, it may be said that ammonia is preferred in this country because of its nominal pressures for a large range of work using condenser pressures varying from those corresponding to 50 to 80 or even 90 deg. F. and suction pressures from —5 to 20 or 30 lbs. gage. Its specific volumes are also nominal, thereby allowing moderate sized suction return

lines and headers. The piston displacement required is convenient, and therefore the cost of the compressor and the space occupied are small. The American engineer feels more at home with such conditions and hence ammonia holds its popularity.

TABLE 10. PROPERTIES OF SATURATED CARBON DIOXIDE.
[Jenkin and Pye—Macintire]

Temp. Deg. F.	Pressure	Vol. of Liquid Cu. Ft. Per Lb.	Specific Volume Cu. Ft.	Heat of Liquid (i')	Latent Heat of Vaporization (r)	Total Heat Above 32° F (i'')
—40	145	.01438	0.609	—38.7	136.3	97.6
—30	174	64	0.495	—33.5	132.1	98.6
—20	214	96	0.408	—28.3	127.4	99.1
—15	234	.01514	0.371	—25.7	125.1	99.4
—10	256	32	0.337	—23.0	122.7	99.7
—5	279	52	0.308	—20.3	120.2	99.9
0	303	70	0.282	—17.7	117.5	99.8
5	329	91	0.257	—14.9	114.8	99.9
10	358	.01613	0.234	—12.2	112.0	99.8
15	389	37	0.215	—9.5	109.0	99.5
20	421	63	0.197	—6.8	105.8	99.0
25	455	89	0.180	—4.0	102.5	98.5
30	490	.01718	0.166	—1.2	98.9	97.7
35	528	48	0.152	1.8	95.1	96.9
40	568	83	0.140	4.7	91.2	95.9
45	610	.01820	0.128	7.6	87.3	94.9
50	653	63	0.117	10.6	83.1	93.7
55	699	.01908	0.107	13.7	78.7	92.4
60	747	53	0.097	16.7	74.1	90.8
65	797	.02000	0.087	19.7	69.0	88.7
70	850		0.078	22.8	63.3	86.1
75	906		0.070	25.9	56.5	82.4
80	964		0.061	28.9	47.1	76.0
85	1029		0.054	32.0	33.0	65.0
88.4	1078		0.049	34.0	0	34.0

It should be said, though, that the carbon dioxide machine is making strides in its applications in the United States. With a better understanding of the properties of CO_2 and the actual cycle of operation, combined with improved designs of the compressor and the remainder of the system, there

TABLE 11. PROPERTIES OF ETHYL CHLORIDE.
Henning.

Temp. Deg. F.	Pressure Lbs. Per Sq. In. Abs.	Vol. of the Liquid Cu. Ft.	Specific Volume Cu. Ft.	B.t.u. Heat of Liquid Above 32° F.	B.t.u. Latent Heat of Vaporization	B.t.u. Total Heat.
—22	2.13	0.0163	34.2	—23.1	193	170
—13	2.80	0.0164	26.5	—19.2	192	172.5
—4	3.63	0.0164	20.9	—15.4	191	175.5
+5	4.63	0.0167	16.7	—11.5	190	178.5
14	5.84	0.0169	13.5	—7.7	188.5	181.0
23	7.28	0.0169	11.0	—3.9	187.5	183.5
32	9.00	0.0170	9.1	0	186	186
41	11.00	0.0172	7.6	3.9	184.5	188
50	13.55	0.0174	6.3	7.7	182.5	190
54.5	14.70	0.0175	5.6	9.5	181.5	191.5
59	16.10	0.0176	5.4	11.5	180.6	192.5
68	19.26	0.0176	4.55	15.4	179.5	194
77	22.90	0.0178	3.90	19.2	176.5	196
86	27.05	0.0178	3.35	23.1	174	197.5
95	31.77	0.0180	2.89	27.0	172	199
104	37.11	0.0182	2.57	30.8	169	200

TABLE 12. PROPERTIES OF SUPERHEATED AMMONIA
Goodenough

Pressure lbs. per sq. in. Abs.	Temp. of Sat. Vapor Deg. F.		Superheat Deg. F 20	40	60	80	100	150	200
		Vol.	14.2	14.9	15.6	16.3	16.9	18.6	20.2
20	—15.9	I	545.2	556.1	566.7	577.1	587.4	612.7	638.0
		Vol.	7.40	7.76	8.12	8.46	8.80	9.64	10.46
40	12.2	I	553.9	565.5	576.8	587.7	598.3	624.6	650.6
		Vol.	5.04	5.30	5.54	5.77	6.01	6.57	7.12
50	30.5	I	559.1	571.2	582.9	594.2	605.2	632.1	658.6
		Vol.	3.85	4.04	4.22	4.40	4.57	5.00	5.42
80	44.5	I	562.8	574.3	587.4	599.0	610.3	637.7	664.6
		Vol.	3.10	3.26	3.41	3.56	3.70	4.05	4.38
100	56.0	I	565.7	578.6	590.9	602.8	614.3	642.2	669.4
		Vol.	2.61	2.74	2.87	2.99	3.11	3.40	3.68
120	65.8	I	568.2	581.4	593.9	605.9	617.6	646.0	673.4
		Vol.	2.25	2.37	2.48	2.58	2.69	2.94	3.17
140	74.5	I	570.2	583.6	596.4	608.7	620.6	649.3	676.9
		Vol.	1.98	2.08	2.18	2.27	2.36	2.58	2.79
160	82.3	I	572.0	589.7	598.6	611.1	623.2	652.1	680.1
		Vol.	1.77	1.86	1.94	2.03	2.11	2.31	2.49
180	89.4	I	573.6	587.5	600.6	613.2	625.4	654.7	682.8
		Vol.	1.59	1.67	1.75	1.83	1.90	2.08	2.25
200	95.9	I	575.0	589.1	602.4	615.2	627.5	657.0	685.4
		Vol.	1.28	1.35	1.41	1.47	1.53	1.68	1.81
250	110.1	I	578.0	592.4	606.1	619.3	631.9	661.9	690.8
		Vol.	1.07	1.13	1.18	1.23	1.28	1.41	1.52
300	122.4	I	580.5	595.3	609.3	622.7	635.5	666.1	695.4

Vol. equals Volume of One lb. in Cubic Feet.
I equals Total Heat in B.t.u. above 32 deg. F.

is no question but that the carbon dioxide cycle has a future. The fear present in the use of ammonia would be avoided as also the danger of fire and explosion.

As regards the *properties of the refrigerants* little is known except in the case of ammonia. There has been little research work on carbon dioxide, sulphur dioxide or ethyl chloride and precise information is lacking. However, most calculations are accurate enough with the information already available. Values are given in Tables* 8, 9, 10 and 11 for the heat of the liquid, the latent heat of vaporization, the specific volume (the volume of one pound) and the temperature and the pressure. With these values most of the practical problems may be solved, as explained in various parts of the text.

*Recently (Dec., 1920), the Bureau of standards' physical properties of ammonia was made public by the American Society of Refrigerating Engineers. These tables embracing the saturated region only have been the continuous work of the Bureau of Standards for a number of years. The results are quite at variance with those of Goodenough, but the author considers that they represent more uniform and concerted work than any other set of tables on ammonia. These tables are probably not final, but they should be accepted as standard until we are warranted in replacing them by others. It should be noted that the ammonia tables are calculated from a zero of heat content from minus 40 deg. F. and therefore no negative values for the heat of the liquid will enter into the calculations. The other tables are calculated according to the *usual* method employing a zero of heat content at 32 deg. F. As the Bureau of Standards has not published the properties of superheated ammonia those of Goodenough are used in Table 12. The reader is warned against the confusion which exists because of the difference in calculation of the properties of refrigerants. Tables 9, 10, 11 and 12 are calculated with 32 deg. F. as the zero of heat content.

CHAPTER VII.

Heat Transfer

Heat transfer is a very broad subject, embracing practically every application of engineering in one way or another. In refrigeration it is studied in its effects as regards the water jacket, the condenser, the expansion coils, and the suction and the discharge pipe lines. Besides these there is the matter of heat leakage through the building walls, floors, and partitions, and through the pipe lines where heat transfer is not desired and which must be insulated. In the absorption machine every part except the aqua pump is designed to permit a transfer of heat. The consideration of the freedom of heat transfer through materials, arranged in different ways with different applications therefore becomes one of primary importance in refrigeration. The subject naturally divides itself into two parts; that of attempting to prevent heat flow, and that of trying to assist heat transfer.

Insulation. An analogy has already been made between flowing water, electricity and heat. Heat cannot run "up hill" or from a colder to a hotter object until it is acted on by mechanical energy. On the contrary, heat has a tendency to flow in the direction of the colder object, to warm this object and finally to establish a condition of equilibrium consisting of an average temperature of all the adjacent bodies. In refrigeration the heat in the atmosphere has a tendency to flow into the ice tank, cold storage rooms or brine cooler or pipes and to raise their temperatures. The result would be

a great loss of capacity, inasmuch as refrigeration would have to be provided in order to carry the live load plus the "leakage" losses due to the conduction of heat from the outside. To reduce this loss to a minimum insulation is resorted to.

In general it may be said that if two adjacent bodies have different temperatures the tendency is for an average temperature to be established. For example the hotter substance tends to become cooler, and the colder one to become warmer. The method producing this temperature equilibrium may be radiation, convection, or conduction. *Radiation* is the method by which the sun's rays reach the earth or by which one receives the sensation of heat from an intense fire perhaps some distance away. It is explained as a vibration of the *ether,*— the substance existing in interplanetary space. *Convection* is the method by which heat is conveyed by a moving substance, like hot air or hot water heating in the factory or the home or by steam from the boiler to the engine, or the brine from the storage rooms to the brine cooler. Finally *conduction* is the manner in which heat is carried along by the particles (molecules) of a solid, as in the case of a hot slicing bar, a hot piston rod, or steam pipe. In the case of refrigeration the heat comes from the atmospheric air as a rule, and passes into the rooms which are refrigerated, or into the insulated (cold) tanks or pipes.

Conductors of Heat. The reason for the *passage of heat* from the air to the refrigerated pipes is self-evident on consideration of the conditions that prevail. Heat passes, like water, from a condition of higher level to one of lower level. The heat in the combustion chamber of a boiler passes through the tubes and shell into the (relatively) cold water in the boiler, and the gaseous products of combustion are lowered in temperature and the water in the boiler is heated and evaporated into steam. In like manner heat from the atmosphere outside the building or from the air inside or the ground underneath the building will leak in. Where we desire to pre-

vent the passage of heat we use a *non-conductor* so placed as to offer a resistance to the passage, similar to the use of magnesia on steam pipes.

It is not, however, practically possible to prevent all losses due to the phenomenon of heat transfer. All that can be done is to use a substance of high resisting power to heat conduction which of itself will not require much volume. Of course, if it were possible to secure a material of great strength as well as resistance to heat transfer then the problem would be different, but brick, concrete, iron and steel are relatively *good conductors* of heat. Therefore, the load due to the dead weights and the storage commodities or the pressure due to the brine or the refrigerating medium must be carried by the appropriate material and the insulation must be added as a necessary extra, very much like the insulation used in steam pipes and steam boilers. The problem, though, is not a simple one. Too much space must not be taken up by the insulating material. The problem is one to be decided from the consideration of the material used, its cost and the value of the space occupied by the insulation. Naturally also the material should be of the same degree of endurance as the rest of the building, and it must be capable of lasting sufficiently long in good condition to make its use economical.

Insulation. From the foregoing it may be seen that the problem narrows down to the following: To conserve refrigeration, insulation of a suitable material must be provided on all sides of the cold-storage room and on the brine and return ammonia pipe lines. Also that perfect insulation is impossible and that the proper amount and the kind of material to be used are dependent on conditions that prevail. These conditions vary with the average summer temperatures, with the kind of cold storage or refrigerating duty performed (and the length of this service) and the cost of a ton of refrigeration.

The insulating material is really a *part* of the building construction, and it must therefore be on a par as far as the quality and the erection are concerned with the rest of the building. It must be capable of withstanding deterioration or rotting, and, in addition, should be free of odor of any sort always. It should be capable of maintaining itself in position (even in the case of a hot fire in its neighborhood) and of not settling in places so as to leave pockets or air spaces. It must be *waterproof* so that moisture may not be collected, as this tends to make it mold and to diminish its ability to insulate, and it must be vermin proof. Finally, the insulator must be of a high degree of perfection and must be capable of reducing the heat losses.

Effect of Moisture. The problem of insulation is much more difficult as far as moisture is concerned than is the case with steam. With steam the tendency is to dry out the covering and to maintain this dryness as long as the temperature of the steam is maintained inside the boiler or vessel. In refrigeration the opposite is true and moisture has the tendency to travel towards the coldest surfaces and to keep collecting as time elapses. In this case also it is possible to have ice form, which may loosen or disintegrate the insulation.

The theory of insulation is somewhat difficult to understand at first. According to this theory there is no material that resists heat perfectly. Remember that insulating for refrigeration purposes is preventing heat from coming inward from the atmosphere to the region maintained at a temperature which is *below the surroundings.* However, it is found that air pockets of extremely small size are an insulation, and any material which has the greatest number of these pockets is the best insulation, but each pocket must be separated from its neighbor. If the pockets are connected, then convection will take place as in the case of the hot air furnace system. From this it would naturally follow that the material that is the best insulator is also the *lightest* in point of *weight,* as a large

amount of its volume is air. This is the reason that substances like mineral wool, corkboard, balsa wood, shavings, sawdust and animal wool have insulation values.

Methods of Securing Insulation. The use of *air spaces* (Fig. 76) naturally follows from what has just been said. However, although under test conditions the air spaces (say 7/8 inch thick) would seem to be efficient, such an insulation does not seem to meet with success. In fact, the reason for the insulating value seems to lie in the effect of the surface resist-

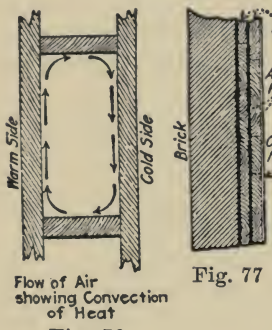


Fig. 76

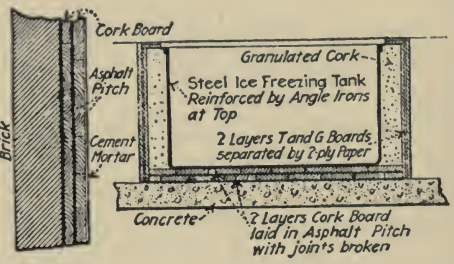

Cork Board
Asphalt Pitch
Brick
Cement Mortar
Granulated Cork
Steel Ice Freezing Tank Reinforced by Angle Irons at Top
2 Layers T and G Boards separated by 2-ply Paper
Concrete
2 Layers Cork Board laid in Asphalt Pitch with joints broken

Fig. 77 Fig. 78

Fig. 76. Heat transfer by convection
Fig. 77. Wall construction using cardboard, asphalt, pitch and cement mortar
Fig. 78. Use of granulated cork for ice tank insulation, corkboard on bottom of tank

ance rather than in any other property. The surface resistance is important in certain cases as in window glass, where it is practically the only resistance to the flow of heat. But in built-up insulation, subject to the accumulation of moisture, etc., it does not seem to be practical.

Sawdust and shavings were used extensively at one time, and still have certain advantages. They are not waterproof nor are they vermin or fire proof. They lend themselves to cheap temporary construction, such as ice storage buildings put up for a short life or for short refrigerating loads. Where

these substances are used the work should be carefully performed, and the shavings or sawdust should be carefully packed and a waterproof paper very carefully secured in place. Sawdust is likely to settle and leave a space in the upper part of the wall, hence shavings, dried before using, are preferred.

Lately, for second-class construction, a large number of *special materials* have appeared in the form of convenient sized boards. They are compressed wood and vegetable fibres so devised that the board may be erected like lumber and nailed into place. Being compressed, it has the characteristics of shavings and has the advantage of more efficient insulation and of being able to retain its position. It is not vermin, fire, or water proof, and thus it would generally be used only in wooden constructions.

Some success has been achieved in the use of *mineral wool,* the product of blast furnace slag or of special mineral earths. These are made up in the form of boards of the standard size, 12 in. by 36 in., and attempts are made to waterproof the material in the process of the manufacture. When so done it is easily erected into place, is fireproof and is a very good insulator. It is used in permanent construction and should maintain itself for years. By far the most successful and popular insulating material is *cork,* obtained from the bark of trees found in Spain and Portugal. This bark has the air cell construction which is necessary for the use we desire of it. For refrigeration use is generally made of the wastage in the manufacture of cork stoppers, etc. The cork particles are compressed and heated so that they adhere together and are made up into boards of standard size or in forms to fit standard pipes and tanks. The cork is inherently waterproof, and in erecting the joints and courses are made up in asphalt pitch or Portland cement mortar. Asphalt is not as satisfactory as mortar because of the possibility of the effect of intense heat—as in the case of fire—loosening the

bond and allowing the insulation to fall. In building construction there is also a final coat, as a rule, of cement plaster which protects the cork board (Fig. 77) and gives a smooth surface suitable for cleaning.

TABLE 13. REPORT OF TESTS CONDUCTED BY THE UNITED STATES BUREAU OF STANDARDS, WASHINGTON, D. C., COVERING THERMAL CONDUCTIVITIES OF VARIOUS INSULATING MATERIALS

Material	Conductivity 24-Hr.Ft.Sq. In. Thick B.t.u.	Density	Nature of Material
Air	4.0	.....	Horiz. layer heated from above radiation
Colorax	5.3	0.064	Fluffy finely divided mineral matter (elim.)
Hair felt	5.9	0.27	
Keystone hair	6.5	0.30	Hairfelt confined between layers building paper
Pure wool	5.8	0.107	Firmly packed
Pure wool	5.8	0.102	Firmly packed
Pure wool	6.3	0.061	Loosely packed
Pure wool	7.0	0.039	Very loosely packed
Cotton wool	7.0	0.100	Firmly packed
Insulite	7.1	0.19	Pressed wood pulp—rigid, fairly strong
Linofelt	7.2	0.18	Vegetable fiber confined between layers paper—soft and flexible
Corkboard (pure)	7.4	0.18	
Eelgrass	7.7	0.25	Inclosed in burlap
Flaxinum	7.9	0.18	Vegetable fibers—firm and flexible
Fibrofelt	7.9	0.18	Vegetable fibers—firm and flexible
Rock cork	8.3	0.33	Prest rock wool with binder—rigid
Balsa wood	8.3	0.12	Very light and soft
Waterproof lith.	9.8	0.27	Rock wood, vegetable fiber and binder, not flexed
Pulp board	10.4	.....	Stiff pasteboard
Air cell ½-inch	10.7	0.14	Corrugated asbestos paper, inclosing air spaces
Air cell 1-inch	11.5	0.14	Corrugated asbestos paper, inclosing air spaces
Asbestos paper	11.8	0.50	Fairly firm but easily broken
Infusorial earth	13.9	0.69	(Block)
Fire felt (sheet)	14.3	0.42	Asbestos sheet coated with cement —rigid
Fire felt (roll)	15.3	0.68	Soft, flexible asbestos
3-ply Regal roofing	16.7	0.88	Flexible tar roofing
Asbestos mill board	20.2	0.97	Prest asbestos, fairly firm, easily broken
Woods—Kiln Dried			
Cypress	16.0	0.46	
White pine	19.0	p.50	
Mahogany	22.0	0.55	
Virginia pine	23.0	0.55	
Oak	24.0	0.61	
Hard maple	27.0	0.71	
Asbestos wood(sanded)	65.0	1.97	Asbestos and cement—very hard and rigid

Special kinds of work also use *granulated* or pulverized cork, as in insulating the sides of ice-making tanks (Fig. 78) and the ends of cylindrical tanks. Usually granulated cork

should be used only where access is readily obtainable for repacking from time to time should settlement occur.

The operating engineer will find a number of methods of calculation of the insulating value of building material, and he will also find some difference in the values reported for the different materials. Elaborate experiments have been made, both with simple materials and with built-up construction. As there are a number of methods of securing the experimental results, some including and others excluding the "skin effect" these variations are accounted for in part. However, the skin effect is of influence on the outside of the insulation only, and for good insulating material of high insulation abil-

TABLE 14. COEFFICIENT OF CONDUCTIVITY OF BUILDING MATERIAL.

Brickwork	5.0
Concrete	5.3 (average)
Wood-fir ⅞" thick	1.0
Asbestos-sheets or boards	0.3 to 0.5
Glass—0.085" thick	24.3
Double window—½" air space	1.10
2" Hollow Tile, plastered	1.0
4" Hollow Tile, plastered	0.6
Mortar	8.0

The figures are the B.t.u. per square foot, per inch thick (unless the thickness is mentioned) per degree difference of temperature per hour.

ity the skin effect may be disregarded without appreciable error.

Neglecting this skin effect—in order to simplify the calculation—the insulating value of the material used is dependent on its conductivity. By this is meant the number of heat units that will pass through it, usually rated for each square foot, one inch thick, per hour or per 24 hours with a difference of temperature on each side of the insulation of 1 deg. F. If this "leakage" is known for one degree difference, then the leakage loss will be proportionately greater for 25, 50, 75 or 100 deg. F. difference in temperature. Likewise if a material one inch thick will give a certain leakage loss, then one 2

or 6 in. will give proportionately less loss. This can be expressed in the formula:

$$Q = A \times K \times (t_{outside} - t_{inside})$$

where A equals the area of the material of the wall or the pipe in square feet, K equals the conductivity of the material, and t equals the temperature in degrees F.

The value of K is obtained from tables of experimental values, as given in Table 13, or (should a built-up section be used) by the following formula:

$$K = \frac{1}{\frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3} + \frac{T_4}{C_4} + \text{etc.}}$$

where T_1, T_2, etc., equal the thickness of each material in inches, and C_1, C_2, etc., represent the conductivity as given in the tables for standard conditions.

Calculations: As an example of the use of the formula, suppose a wall composed of 8 in. of brick, 2 in. of corkboard and ½ in. of mortar is exposed to temperatures of 70 deg. and zero deg. F. on the two sides and it is desired to find the leakage per twenty-four hours for a total surface of 700 sq.ft. The value of K becomes (see values in the table)

$$K = \frac{1}{\frac{8}{5.0} + \frac{2}{0.31} + \frac{0.5}{8.0}} = 0.122 \textit{ B.t.u. per hour}$$

and the leakage is

$$Q = 700 \times 0.122 \times (70 - 0) = 5978 \text{ B.t.u. per hour.}$$
$$= 5978 \times 24 = 143{,}470 \text{ B.t.u. per 24 hrs.*}$$

The practical engineer does not care for great refinements in the calculation of insulation any more than for similar details in other branches of the work. He desires a quick answer to the problem with the least calculation possible. It is not recommended for these reasons that he consider much more

*For skin effect increase the denomination by 0.77 for *each inside* surface and by 0.25 for an *outside* surface.

TABLE 15

STANDARD BRINE THICKNESS

Pipe Size inches	Trans. in B.t.u. per lin. ft. per deg. dif. in temp. for 24 hours	Pipe Size inches	Trans. in B.t.u. per lin. ft. per deg. dif. in temp. for 24 hours
½	3.37	4½	5.93
¾	3.53	5	6.75
1	3.73	6	7.03
1¼	3.87	7	8.49
1½	3.96	8	8.72
2	4.44	9	9.04
2½	4.84	10	10.01
3	5.20	12	11.46
3½	5.46	14	12.36
4	6.21	16	13.80

SPECIAL THICK BRINE

Pipe Size inches	Trans. in B.t.u. per lin. ft. per deg. dif. in temp. for 24 hours	Pipe Size inches	Trans. in B.t.u. per lin. ft. per deg. dif. in temp. for 24 hours
½	2.92	4½	5.02
¾	3.17	5	5.61
1	3.22	6	5.98
1¼	3.43	7	6.54
1½	3.67	8	7.10
2	3.90	9	7.67
2½	4.40	10	8.30
3	4.83	12	9.43
3½	4.78	14	10.13
4	5.31	16	11.26

ROOMS OR BUILDINGS

Thicknesses of Corkboard Recommended for

Range of Temperatures	Walls	Ceilings	Floors On Ground	Floors Above Ground	Roofs
Below −15° F.	8″	8″	7″	8″	9″
−15° F. to −5° F.	7″	7″	6″	7″	8″
−5° F. to 10° F.	6″	6″	5″	6″	7″
10° F. to 25° F.	5″	5″	4″	5″	6″
25° F. to 40° F.	4″	4″	3″	4″	5″
40° F. to 50° F.	3″	3″	2″	3″	4″
50° F. to 65° F.	2″	2″	...	2″	3″
60° F. & above.	...	...	...	...	2″

TABLE 15—(*Continued*)

FREEZING TANKS

Thicknesses of Corkboard Recommended for

Bottoms		Temperatures	Thickness of Corkboard
If placed on foundation laid on ground.		—20° to —5° F.	Eight inches
Minimum	5″	—5° to +5° F.	Six inches
Preferably	6″	5° to 20° F.	Five inches
		20° to 35° F.	Four inches
		35° to 45° F.	Three inches
		45° and above	Two inches

CYLINDRICAL COOLERS, TANKS AND FILTERS FOR COLD LIQUIDS

Thicknesses of Corkboard Recommended for

Range of Temperatures	Sides, top
Below 0° F.	6°
0° F. to 10° F.	5″
10° F. to 25° F.	4″
25° F. to 45° F.	3″
45° F. to 55° F.	2″
55° F. & above	1½″

than the effect of the conduction of the material used except in special cases where the effect of surface resistance is a large percentage of the total resistance of the heat transfer. For example a piece of glass has hardly any resistance to conduction, and yet there is a "skin" resisting effect which serves the same end. This surface resistance applies to all materials, and to practically every case of heat transfer, as also in the cases of steam boilers, steam condensers, ammonia condensers, etc.

Whereas the skin effect would tend to reduce the value of K and Q, yet the factor which is important is the perfection of the erection, which is subject to wide variations. Unless the joints are carefully made and air tight, moisture will get in and freeze, causing the ultimate deterioration of the insulation. This is probably the factor most responsible for the fact that the laboratory results and the practical results do not agree, and we are justified in adding a liberal factor of safety to insure good operating results. For these reasons

and for simplicity of calculation it is recommended that use be made of the method of calculation given.

As regards the maximum temperature to be used in the calculation of leakage losses, it is customary to assume an average maximum temperature during the period of peak refrigerating loads. Some idea of average temperatures in the United States may be obtained from the map, Fig. 79. It is not recommended that the highest local temperatures be

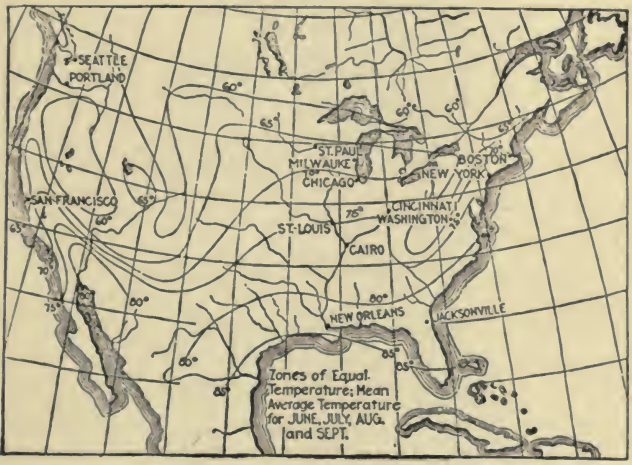


Fig. 79. Map of the United States showing average summer temperature [for use in calculating insulation losses.]

taken, but the average temperature experienced during the twenty-four hours for conditions that are likely to prevail for a period of days, say a week or more. Of course, the actual choice of the thickness of the insulation for any particular case will be decided by the relative costs of the insulation itself, and the book value of a ton of refrigeration. If the time of operation, days per year, is small, one cannot economically install heavy first cost construction. It would be a poor investment. Typical examples of best insulating practice are

shown in the cuts, Figs. 80 to 84, inclusive. Fig. 85 shows a modern freezer cold storage warehouse. It will be noted that very thick insulation is carried on the walls and floors.

Table 15 is added in order to show thicknesses recommended by insulation companies for different cases. In general it can be said that one inch thickness of corkboard should be used

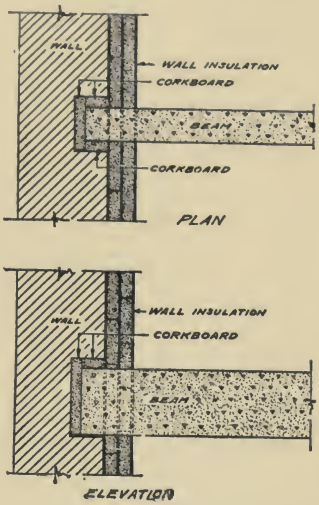


Fig. 80. Typical first class wall insulation [note method used in insulation at the floor level]

for every 10 to 12 degrees difference of temperature to be carried.

In the matter of the heat transfer for ammonia piping the conditions which prevail are much like those existing in steam boilers and steam condensers. The steam boiler is affected by soot and air film on the fire side and scale and surface water *film* on the water side of the tube. The steam condenser is affected by scale on the water side and air film on the steam

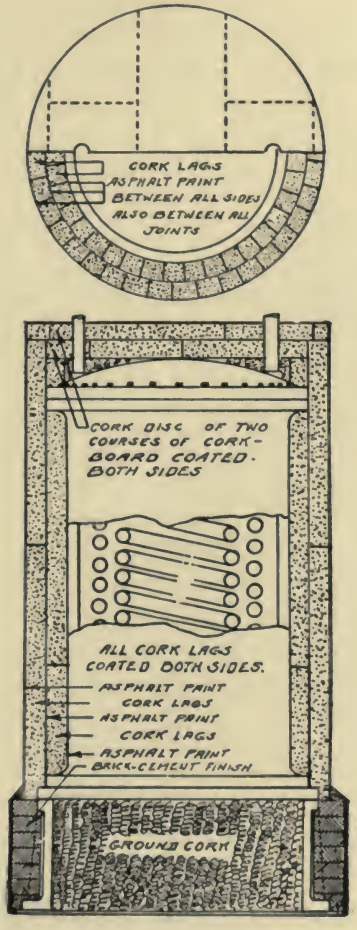


Fig. 81. Typical insulation for use in brine tanks

side. If it were not for scale or other non-conducting coatings on the heat transfer surfaces the same amount of work could be done with a small fraction of the surfaces usually needed, under existing conditions.

In refrigeration the condenser and the other refrigerating surfaces have the same troubles to contend against with the additional difficulty of the accumulation of ice (frost) on the low temperature piping. Frequently also there are designs of nominal efficiency (necessarily so, in many cases) which do not allow the gas formed during the boiling action to be car-

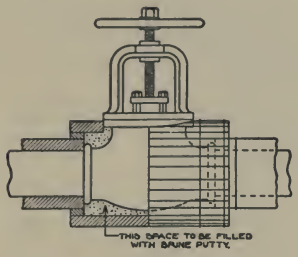


Fig. 82. Typical insulation —Brine globe valve

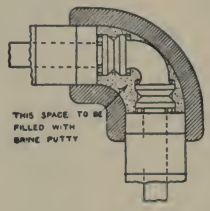


Fig. 83. Typical insulation—Ammonia flanged elbow

ried away quickly, and (in condensers) there is not the quick removal of the condensate which will allow the compressor gas ready access to the condenser surfaces.

One factor of importance in all heat transfer surfaces is that of the agitation of the surrounding fluid. The best in-

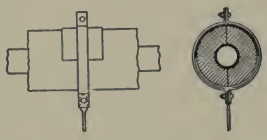


Fig. 84. Typical insulation—brine pipe and hanger

sulating material is "still" air. In refrigeration, in steam boilers and steam condensers there seems to be a tendency for the collection of inert gases and the formation of an air film to the pipe surfaces which sticks to the surface and greatly increases the resistance to the passage of heat through the

walls of the pipes or tubes. Anything that can be done to "scrape" away this film by creating a velocity of flow in the medium will increase the coefficient of heat conduction or the capacity of the piping. This is true in the case of the radiator, the bunker coil, the ice-making tank or the cooling tower.

Expansion piping is also allowed to become covered with

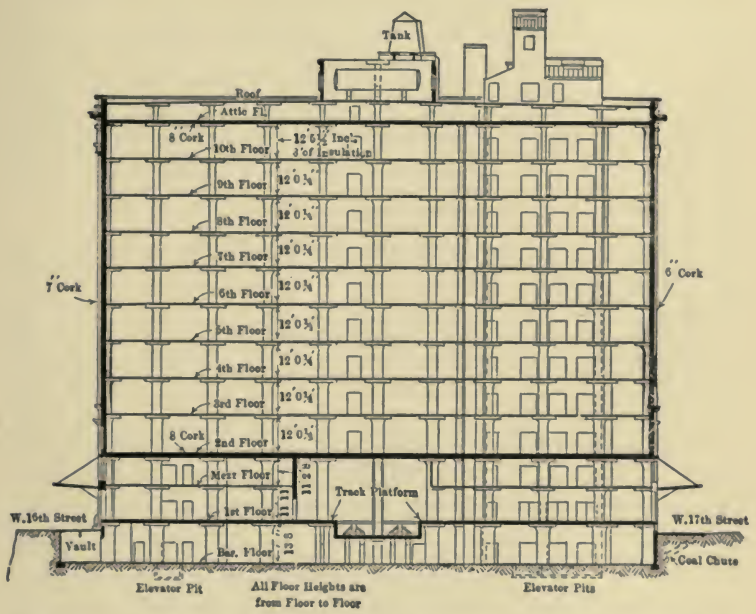


Fig. 85. Cross section, Merchants Refrigerating Co., Cold Storage Warehouse, showing Freezer Insulation

frost. Some engineers believe that as the diameter increases with the addition of frost the increased surface will make up for the decreased rate of heat conduction. One has only to try to get capacity with the expansion piping loaded with frost in order to see the fallacy in such an idea. Frost must be removed, either by scraping or by the usual methods of defrosting. One method used with success is to allow brine to

trickle down the pipes, arranged as wall piping in stands; this brine will keep the surfaces wetted and at the same time will prevent the accumulation of frost. Whereas the coefficient of conduction is increased, the extra labor involved in attending to the brine has prevented its general use.

From the foregoing it can be seen that the values for heat transfer in refrigeration piping are very variable. In consequence such cannot be stated with much surety, and the best that can be offered are average results—or those that have been found to be successful under a variety of different installations. A few of these values of "k" are as follows; in *B.t.u.* per *square foot* of pipe surface per *degree difference* of temperature per *hour*:

Freezing tanks	
Old style feed (expansion valve on each coil)	12 to 15 B.t.u.
Flooded	20 to 30 B.t.u.
Ammonia condensers	
Submerged type (practically obsolete at present)	30 to 40 B.t.u.
Atmospheric type (old style, parallel flow)	60 to 65 B.t.u.
Double pipe	150 to 200 B.t.u.
Bleeder (drip pipe, counter flow)	125 to 200 B.t.u.
Flooded type*	140 B.t.u.

Engineers sometime prefer to have the condenser surface stated in terms of the area per ton of refrigeration, and average results are as follows, remembering that the condenser has to remove about 260 B.t.u. per ton. The shell and tube requires 12 to 20 sq. ft., the shell and coil 16 sq. ft., the double pipe 6 to 10 sq. ft., the atmospheric 8 to 10 sq. ft. and the flooded 4 to 8 sq. ft. per ton of refrigeration.

Baudalot coolers (counter flow, atmospheric type)	
Milk coolers	75 B.t.u.
Cream coolers	60 B.t.u.
Oil coolers	10 B.t.u.
Brine coolers	
Shell and tube	90 to 100 B.t.u.
Cooling Coils	
Brine to unagitated air	2 to 2¼ B.t.u.
Direct expansion to unagitated air	1½ to 2 B.t.u.

* Numerous claims for better results than those here given for the flooded condenser have been advanced, but these are disputed.

The action of forced circulation of the air increases the heat transfer, and may be taken as 1½ to 2½ times the values given above. Frost on the pipes reduces the value of "k," and one inch of frost will reduce k by about one-fourth.

CHAPTER VIII

Piping and Piping Calculations

The expansion piping of the refrigerating system has its counterpart in the boilers of the steam plant. The boilers absorb heat from the hot gases produced by the combustion of the fuel. While with the single boiler the fireman has a ready means of telling about the output, in a battery of boilers there is no ready means,—except possibly by checking up the amount of coal burned with the flue temperature and the CO_2 analysis—of telling whether one or more boilers are not loafing. The steam meter, which gives immediately the response of any boiler to the action of draft and stoking by registering the steam output in pounds per unit of time, is an accurate register of a boiler's output.

In like manner if the expansion piping or refrigerating coils are "laying down" on the job, it is even harder to tell just where this is taking place, aside from the fact that results are not being obtained. The operator may be holding the suction pressure right, the condenser and all the high pressure side may be apparently in good shape, but the ammonia will seem to disappear from the receiver and the required temperatures will not be maintained. The trouble in such cases is that the ammonia is not being evaporated as it should. It is the *boiling of liquid ammonia at the proper pressure* that gives useful refrigeration. The improper pressure, the presence of oil or scale or water on the inside of the pipe and scale and ice on the outside decreases the ability of the ex-

pansion piping to function—and the result is reduced capacity—sometimes very much reduced capacity. With no flow meter the operator can not readily check the several parts of the system. If results are not obtained practically the only course is to check over the possibilities for trouble and do such overhauling as seems necessary and desirable.

Operating Pipe Troubles. Refrigerating systems have *troubles* which arise somewhat in the same manner as those

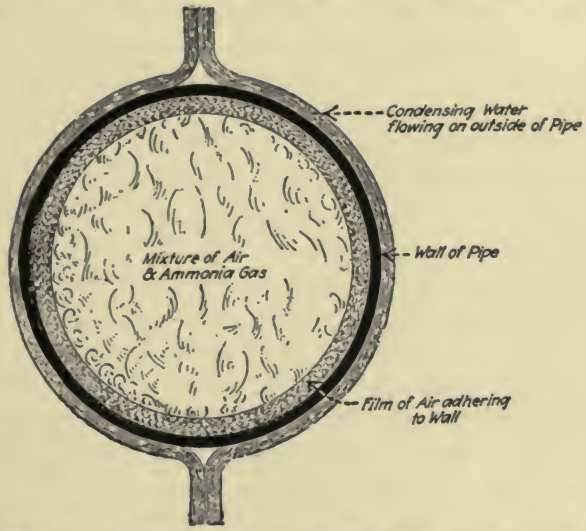


Fig. 86. Looking into a condensed pipe. Note factors tending to reduce heat transfer

in a boiler plant. The system may accumulate air, on account of repair work, or with the low pressure side operating under a vacuum, or by carelessness in the initial charge of the system. Then moisture may get into the expansion piping when the engineer is blowing out the coils with steam in order to remove an accumulation of oil, or even by being carried into the system by condensing on the piston rod. The oil, of course, gets into the coils by being carried out of the cylinder

by the discharge gas. Let us see how these impurities affect operation and the production of refrigeration.

The effect of *air* in the steam condenser is to increase the back pressure on the steam engine or the steam turbine. This decreases the economy of operation. Air or some other inert gas in the refrigerating system collects in the condenser, where it increases the head pressure over and above that which should exist due simply to its own vapor tension. The head pressure may be 10, 20, 50, or more pounds greater than it should be, and the horse power required to produce a ton of refrigeration will be correspondingly increased.

In addition the air collects in the condenser, first forming an insulating film about the condenser pipes, (Fig. 86) and then—as it accumulates—fills the condenser. It is a mistake to think that air or an inert gas collects in any particular part of the condenser. All gases tend to diffuse and become uniformly mixed. The only way to separate the ammonia from the gas is to condense it. This is done partly in the process of purging when the cooling water is showered over the pipes while the compressor is not pumping on that particular stand; partly by allowing the gas in the upper part of the condenser to bleed out, making it less rich in inert gas in the process.

Moisture in Expansion Piping. Air in the system is thus seen to affect the *head pressure* and therefore to increase the horse power needed per ton, but the accumulation of water in the low pressure piping has a yet worse effect. It may prevent obtaining the required temperature in the refrigerating coils. Moisture *will* get into the system in one way or another. If it gets in by way of the condenser, it mixes with the anhydrous ammonia in the receiver and finally passes through the expansion valve into the refrigerating coils. As the boiling pressure of the ammonia (the back pressure) is relatively high, there is an almost negligible tendency of the water in the coils to evaporate. The water remains in the low pressure

side and forms first a strong aqua ammonia, and then (as the water increases) a less strong aqua.

Aqua causes the cooling coils to behave as if *dead.* The boiling action is decreased, or the temperature required cannot be obtained. To get the results required, a lower suction pressure must be carried than would be required if the ammonia were free from water. The result of a lower suction pressure for refrigerating compressors has already been given considerable emphasis, increasing as it does, the horse power per ton of refrigeration and decreasing the capacity of the

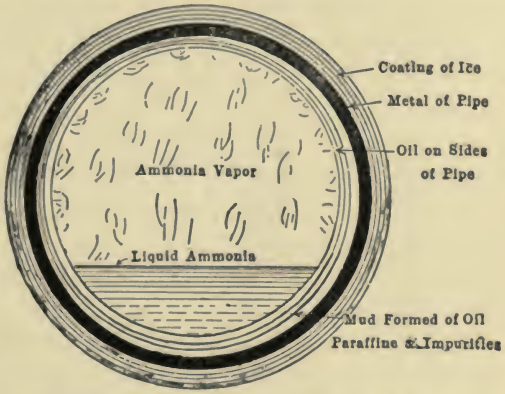


Fig. 87. Section of a cooling coil when Functioning

machine. The net result may be noted in the table, which shows the boiling temperature of aqua ammonia under various pressures and per cents of water. For example, at 20 lbs. gage pressure anhydrous ammonia will boil at 5.5 deg. F., but if 10 per cent. of water is present the temperature of boiling of the solution (at 20 lbs.) will be 15 deg. F. Therefore (see table) the pressure of the suction will have to be depressed to 12 or 13 lbs. to give the same boiling temperature with 10 per cent of water present in the ammonia as with pure anhydrous ammonia boiling at 20 lbs.

Oil in Expansion Piping. No refrigerating system can be kept entirely *free from oil.* The vertical enclosed type compressors are splash lubricated, and the horizontal high speed machines must have copious amounts of oil or trouble will occur. Even the old style, slow speed compressors carry over considerable oil which is supposed to be caught in the oil separator. However, it is well known that no separator can remove all the oil from the discharge gas. Some is volatilized and finally condensed in the condenser with the liquid ammonia, while some other portion passes the separator in the liquid form. In fact it is safe to assume that from 2 to 5 per cent of the oil will get into the condenser and ultimately collect* in the low spots (if there are any such) of the direct

TABLE 16.

Suction Pressure Lb. Gage.	Boiling Temp. Saturated Ammonia Deg. F.	Boiling temperature. 5 per Cent. Moisture Deg. F.	10 per Cent. Moisture Deg. F.	15 per Cent. Moisture Deg. F.
30	16.6	21	26	31
25	11.3	16	21	28
20	5.5	11	15	21
15	—1.0	4	9	13
10	—8.4	—2	1	6
5	—17.0	—11	—8	—3

expansion piping. It may mix with impurities or dirt and form a thick mud, or it may solidify at the temperatures carried in the system.

The net result of oil in the expansion piping is for reduced efficiency of the coil surfaces. The oil or grease will form an insulating covering, which increases the resistance to the flow of heat through the pipe. Ammonia is evaporated by being in an *atmosphere* which is hotter than the boiling temperature of the ammonia at the *pressure* being maintained in the coils. Therefore heat passes into the pipe coils, and the ammonia is boiled away to make room for more liquid passing

* A drain from the bottom of the liquid receiver will decrease this danger.

into the coils through the expansion (pressure reducing) valve. In consequence of heat being used up to boil the ammonia, the atmosphere or brine is cooled. It is desired to get as many heat units past the walls for every square foot of pipe surface as possible under the circumstances, but an oil film 1/100 of an inch thick is as bad as a boiler scale 1/10 of an inch, or a steel plate 10 inches thick.

Unlike steam surface condensers, which are usually of brass and some 0.05 in. thick, refrigerating expansion piping is usually 1¼ and 2 in. normal diameter and the walls are usually extra heavy. The outside surface is frequently heavily coated with ice; the net result is a great reduction in the unit heat transference, (see Fig. 87). If a pipe will not allow sufficient heat to pass then more surface will be required or a greater temperature difference will have to be carried. Either of these two alternatives is commercially unbearable and so the piping must be kept in as clean a condition as possible, just as in boiler and steam condenser practice.

Maintaining Piping Clean. The best possible arrangement of piping to keep the low pressure side free from oil or water is to arrange all piping so that it will drain and so that it will contain no pockets. At each low point there should be installed a suitable trap which may be opened to the atmosphere or piped to the regenerator apparatus. Although semi-solid oil will not flow readily, and may remain on the sides of the pipe, yet ease in cleaning is obtained and the oil may be made to flow when desired by passing some of the hot gas from the compressor directly into the expansion coils. More trouble is encountered with the low pressure side, perhaps, than with any other part of the system. This is mainly due to the fact that the low pressure side is not so well understood. And yet it certainly is not more difficult to understand than is the steam boiler, the steam surface condenser, the closed feed water heater or the economizer. Each of these devices is a heat exchange apparatus, and the heat transfer is

made possible by having *metal* contact with the two substances. But to transfer heat there must be a temperature *difference* on the two sides of the metal—a temperature difference in the two substances, the ammonia and the brine, or the ammonia and the atmosphere of the room to be cooled.

Suction Pressure. As has been pointed out before, the ammonia in the refrigerating coils must be maintained at the proper pressure for satisfactory service. Why does not a drum of ammonia provide refrigeration and become frosted with a thick layer of frost? Why does not the liquid receiver normally become frosted and tend to refrigerate the compressor room; or the liquid pipe lines refrigerate the air through which it passes in going to the different locations of the expansion valves? Because the pressure in these pipes or drums is too great, and therefore the temperature corresponding (a fact due to the laws of saturated vapors) is greater than or equal to the temperature of the air in the engine room or passage way. To have an evaporation of the ammonia its boiling temperature must be below the temperature of the substance on the other side of the container. Heat must flow into the expansion coils from the outside. The point in question is brought out by every day operation of the plant. If a sharp freezer and a cooler are on the same system and a pressure of 15 lbs. gage is carried, the ammonia will become "lost" and the receiver will become empty. Investigation will show that the sharp freezer coils are flooded because the room was too cold to evaporate the ammonia and it accumulated in these sharp freezer coils. The remedy would be in reducing the back pressure to 10 lbs. or 5 lbs. gage, in which case the boiling temperature of the ammonia would drop to —8.5 deg. and —17.0 deg. F. respectively, and, as the sharp freezer is maintained at zero degrees, boiling action would be stimulated and refrigeration would be maintained. The whole process is expressed by the formula:

Refrigeration in B.t.u. = *area of pipe in sq. ft. times the coefficient of heat transmission times the temperature difference on the two sides of the pipe.*

From this formula it would seem that to secure refrigeration in moderate amounts it is necessary to have either extensive piping or a large temperature difference. For example, a ton of refrigeration can be obtained by the following:

a—400 sq. ft. × 2.0 × 15 = 12,000 B.t.u. per hour.
b—200 sq. ft. × 2.0 × 30 = 12,000 B.t.u. per hour.

where in each case the coefficient of heat transmission is taken as being equal to 2.0. In *b* only one-half the piping is needed as compared with *a*, but the temperature difference is twice as great, thereby requiring about 10 lbs. suction pressure, as compared with the case *a* which requires about 20 lbs. Likewise if 800 sq. ft. were used then only 7½ deg. F. temperature would be required with a correspondingly higher suction pressure. The higher the suction pressure the greater the number of pounds of ammonia handled, and therefore the greater the efficiency of operation with respect to the power input.

Calculation of Piping. As regards the possible demands on the refrigerating pipes it is well known that there are the so-called *live loads* and the *leakages*. These loads vary with each design and location, and may consist of the heat generated by the electric lights and the animal heat of the workmen, the mechanical equivalent of the heat of the fans or other machinery, the heat due to the chemical energy of decay, the sensible and latent heats (if the goods are frozen) of the new goods brought into storage, the liquefaction and freezing of the steam given off by the goods (as in the chill room of a packing house) and finally the leakage of heat from the outside atmosphere which is, of course, nearly always at an amount considerably above the temperature of the cold storage rooms.

With these varied conditions of operation—of refrigerating load and of heat transmission per square foot of pipe surface—it is naturally difficult to design the expansion piping with economy. The main consideration on the part of most engineers is to get enough surface so as to be sure to carry the load readily *under any and all conditions of operation.* The tendency has been to play safe, and to be sure that plenty of pipe surface is provided so that after a term of years of service with even indifferent care and attention the machine may still give the contract load. The old style can-ice plant was laid out for a heat transfer of about 12 to 15 B.t.u. per sq. ft. per degree difference of temperature and those engineers who became familiar with these conditions of design cannot become used to the new plants giving 25, 30 or higher values for "k." The thought is ever in their minds, "Is it possible to get full capacity of ice from the ice tank after the pipes have become fouled with an accumulation of four or five years of oil and scale?" This point of view is the wrong one, and time will see a decided change. The refrigerating machine and piping will be sold with the understanding that these will be maintained in an efficient operating condition. It will be remembered how heavy the copper used to be in electric machinery, and how steam turbines had better steam economy at 25 per cent overload than at full capacity. The reason for these cases was that the designers were not sure of their machines when operating *under industrial conditions* in the field.

The reason for emphasizing so particularly this matter of the expansion piping is to call attention to the difficulties in the way of the engineer who is planning a new system, and also to make clear the reason for the great variation in the amount of piping one notices from time to time. Under good operating conditions there is no question but that relatively small surfaces may be used, but otherwise there must be sufficient allowance for the decrease in unit output.

Example. As an example of the method to be used in the calculation of a typical problem consider the example of a cold storage room having 2,200 sq. ft. of wall surface, and with a leakage coefficient (k) = 0.07 B.t.u. The storage room is designed for 34 deg. F. and for an average of 70 deg. F. outside temperature. The live load consists of 50,000 lbs. per day of goods at a temperature of 65 degrees and of 0.8 specific heat, and of 900 B.t.u. per hour for 8 hrs. as the heat equivalent of two workmen and the lights used in the room. The problem is to find the piping required for 24 hours operation.

Leakage loss—$2{,}200 \times 0.07 + (70\text{-}34) =$ .....5,544 B.t.u. per hr.

Average goods load—$\frac{50{,}000}{24} \times 0.8 \times (65—34) =$ ...51,700 B.t.u. per hr.

Average mechanical load—$\frac{900 \times 8}{24} =$ ..... 300 B.t.u. per hr.

Air leaks during loading (estimated)= .....10,000 B.t.u. per hr.

Total= .....67,544 B.t.u. per hr.

Piping area required$= \frac{67{,}544}{2 \times 20} = 1690$ square feet

$= 1690 \times 1.6 = 2{,}704$ linear feet of 2 in. pipe

In the problem an estimate has to be made of the loss due to the opening of the cold storage doors. This amount depends on the personal equation of the workmen and whether ante-rooms are used. The bringing in of a large amount of relatively hot goods would raise the room temperature, but this point has been neglected because it has been assumed that 34 deg. F. is the constant temperature during the period under calculation. Also the value of $k = 2.0$ has been taken for direct expansion, and a temperature difference on the two sides of the pipes of $34 - 14 = 20$ deg. F. The calculations do not account for the condensation and freezing of water vapor on the two pipes, as such figures would be relatively small in this case. The problem is given to show the factors entering into a calculation, and the results reached. The leakage through the walls may be arrived at by a method similar to that given in the chapter on heat transfer (Chapter VII) for the type of construction of the walls and floors used in the individual case.

The Liquid Line. The liquid line seldom gives trouble, and yet there are a few points about it which are worthy of study. The behavior of the liquid ammonia in the liquid line (as regards the flow of the liquid from the receiver to the expansion valve) is the same as the flow of water from the air chamber of a pump to the discharge nozzle. However, both the lift and the velocity of the ammonia must be nominal in order to get good results.

In nearly every case it does not much matter what drop of pressure there is before reaching the expansion valve providing there is no tendency for vaporization to occur in the pipe due to too great a pressure drop of the liquid ammonia. With a condenser pressure of 150 lbs. it is possible to raise the liquid to a height of 540 ft. (an amount much greater than water could be raised under the same pressure because of the lower density of liquid ammonia). It is possible to locate the expansion valve in any part of the usual type of building unless the friction loss in the pipe is excessive.

As a rule the velocity permitted in the ammonia feed line is from 6 to 10 ft. per second. Knowing the tonnage required, the number of pounds of ammonia needed may be calculated. The volume of this weight may then be found by referring to Table 8 in Chapter VI under the heading of the specific volume, in cubic feet per pound. If the volume in cubic inches of the liquid passing any cross section per second is known, then divide by the velocity desired (72 or 120 in.) and the quotient will be the area cross section of the pipe required. The method is exactly the same as would be used in the case of a water or any kind of liquid pipe.

Arrangement of Expansion Valve. The expansion valve should be carefully located. Of course the valve may feed any length of pipe but naturally the surface which will conduct to the ammonia the heat necessary to boil it should be sufficient for the liquid feed being supplied. Too short a pipe might fail to give enough releasing area to vaporize the am-

monia and the suction return might be very wet. In like manner too great a length of pipe would cause choking of the pipe and inefficient service. In other words the length of direct expansion piping which may be controlled by a single expansion valve is limited to its ability to free itself of the ammonia gas that is boiled out. In general it may be said that the practical maximum which may be used is:

1,100 feet of 1 inch pipe.
1,300 feet of 1¼ inch pipe.
1,600 feet of 1½ inch pipe.
1,900 feet of 2 inch pipe.

With brine pipe the case is somewhat different, as brine is not subject to bad effects from gas formation in the pipe. The amount of pipe per coil is independent of the size of the pipe and varies with the operating conditions from 100 to 400 ft.

Piping Ratios. Table 17 will be found convenient for calculations or for checking. Various room temperatures and boiling temperatures of the ammonia are used, giving a variety of pipe lengths per ton of refrigeration. As a general rule (except for short seasonal operation) it is wiser to have high suction pressure and relatively large piping ratios; in other words to have an economical operation rather than a low initial cost. For direct expansion piping both 1¼ and 2 inch pipe is given, but for brine only 1¼ inch. Should it be desired to find the length of pipe for any other size, say for 2 inch pipe,—using brine—divide the length of pipe as given in the table by the ratio of the external areas (per unit length) of the two pipes. For example, should it be desired to find the length of 2 inch pipe per ton for a room to be held at 20 deg. F. and with a suction pressure of 10 lbs. gage pressure, then, take from the table the value 540 ft. and divide by $\frac{2.3}{1.6}$ (the ratio of the external areas per linear foot) and the answer is 378 feet of 2 inch brine pipe.

As a convenience the curves in Fig. 88 are given. This set of curves is very useful for approximate work and for pre-

Table 17. Lineal Feet of Pipe Per Ton of Refrigeration to Cool Air.

This table is based on a coefficient of heat transmission (K) of 2.5. If any other value of K is desired multiply by 2.5 and divide by value preferred. If no frost accumulates on pipe multiply lineal feet by 0.6.

Suction pressure, lb. gage		45	40	36	30	27½	26½	22	19	15.7	12	10	7	4.6	1.3	—2.7
Temp. ammonia, deg. F.		30	26	22	17	14	12	8	4	0	—5	—8	—13	—18	—25	— 35
Temp. brine, deg. F.		38	34	30	25	22	20	16	12	8	3	0	— 5	—10	—19	— 30
	Kind of Pipe															
Room Temperature 65 deg. F.	2 in. d.e.	220	198	179	160	151	145	135	126	119	110	106	99	93	86	77
	1¼ in. d.e.	314	282	256	229	216	208	193	180	170	157	151	141	133	123	110
	1¼ in. br.	400	349	309	270	252	240	221	204	190	178	167	155	144	129	114
Room Temperature 60 deg. F.	2 in. d.e.	257	227	203	179	168	161	148	138	129	119	114	106	99	91	81
	1¼ in. d.e.	368	324	290	256	239	229	212	197	184	170	162	151	141	130	116
	1¼ in. br.	491	416	360	309	285	270	246	225	208	190	180	167	155	137	120
Room Temperature 55 deg. F.	2 in. d.e.	308	266	234	203	188	179	164	151	140	129	122	113	106	96	86
	1¼ in. d.e.	440	380	335	290	269	256	244	216	200	184	175	163	152	137	123
	1¼ in. br.	635	515	433	360	328	309	277	251	230	208	197	180	167	146	127
Room Temperature 50 deg. F.	2 in. d.e.	385	321	275	234	214	203	184	168	154	140	133	122	114	103	91
	1¼ in. d.e.	550	459	393	337	306	290	264	240	220	200	193	175	163	147	130
	1¼ in. br.	900	675	540	432	386	360	318	284	258	230	216	197	180	157	135
Room Temperature 45 deg. F.	2 in. d.e.	514	406	335	275	249	234	208	188	171	154	145	133	123	110	96
	1¼ in. d.e.	735	580	478	393	356	334	297	269	244	220	207	190	176	157	137
	1¼ in. br.	1540	982	720	540	470	432	373	327	292	258	240	216	197	169	144
Room Temperature 40 deg. F.	2 in. d.e.	770	550	428	335	297	275	241	214	193	171	161	145	133	119	103
	1¼ in. d.e.	1110	785	612	478	425	393	344	306	276	244	232	207	190	170	147
	1¼ in. br.	5400	1800	1080	780	600	540	450	386	338	292	270	240	216	184	154
Room Temperature 35 deg. F.	2 in. d.e.	1540	856	592	427	366	334	286	248	220	192	179	160	145	129	110
	1¼ in. d.e.	2220	1220	845	610	523	477	409	355	314	274	256	229	207	184	157
	1¼ in. br.			2160	1080	830	720	569	470	400	337	308	270	240	200	166
Room Temperature 30 deg. F.	2 in. d.e.		1925	962	592	481	427	350	397	256	220	202	179	160	140	119
	1¼ in. d.e.				845	688	610	500	424	366	314	288	256	229	200	170
	1¼ in. br.				2160	1350	1080	772	600	490	400	360	308	270	220	180
Room Temperature 25 deg. F.	2 in. d.e.			2566	962	700	592	452	367	308	256	233	202	179	154	129
	1¼ in. d.e.				1375	1001	846	645	525	440	366	233	288	256	220	185
	1¼ in. br.						2160	1200	830	636	410	432	360	308	246	196
Room Temperature 20 deg. F.	2 in. d.e.				2566	1283	962	642	482	385	308	275	233	202	171	140
	1¼ in. d.e.						1375	917	689	550	440	393	333	289	244	200
	1¼ in. br.							2700	1350	900	636	540	432	360	277	216
Room Temperature 15 deg. F.	2 in. d.e.					7700	2566	1100	700	514	385	334	275	233	193	154
	1¼ in. d.e.								1000	734	550	477	293	333	276	220
	1¼ in. br.								3600	1543	900	520	540	432	318	240
Room Temperature 0 deg. F.	2 in. d.e.									7700	1540	962	592	527	308	220
	1¼ in. d.e.											1375	846	754	441	315
	1¼ in. br.												2160	1080	569	360
Room Temperature —10 deg. F.	2 in. d.e.												2570	962	514	308
	1¼ in. d.e.													1375	735	441
	1¼ in. br.														1200	540

d.e. means direct expansion.
br. means brine.
The values for piping above are for natural draft circulation of air. If forced circulation, multiply by 0.5 to 0.6.

liminary estimates. It is possible, moreover, to get fairly good results from them provided that the conditions found in the particular cases are not abnormal.

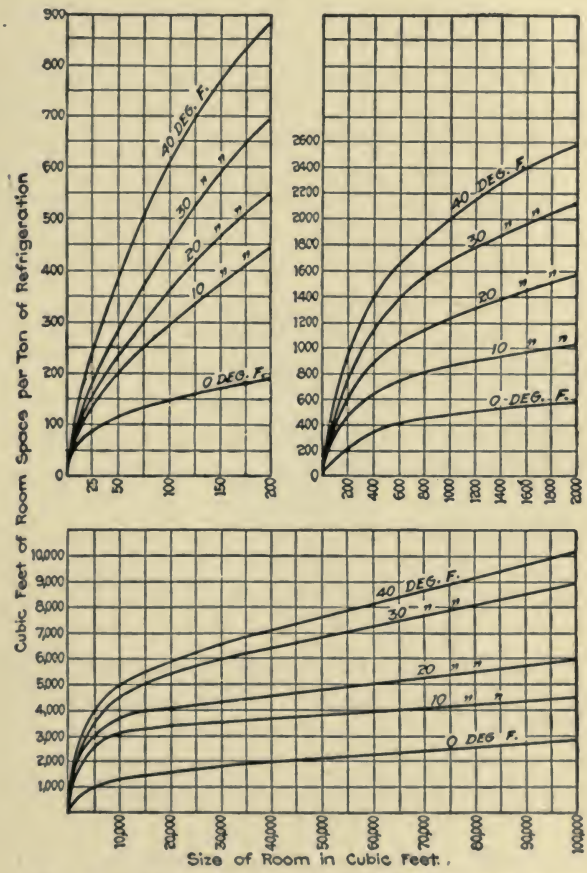


Fig. 88. Diagram showing cubic feet storage space per ton of refrigeration for various sized rooms

Pipe Sizes. All steam engineers understand the importance of the proper pipe sizes and connections. With steam it is necessary to prevent excessive pressure drop as well as to

keep the headers and pipers drained and free from the possibility of water hammer. The pressure drop from the boiler to the engine or turbine results in some loss in efficiency and power. Likewise on the condenser side it means the same thing to a much greater degree.

In refrigeration the same general conditions are true, especially as applied to the suction line. The ideal condition is to have the same cylinder suction pressure (the actual pressure during the suction stroke) as the pressure of evaporation in the expansion coils. This condition is an impossibility because there must be a pressure drop to make the gas flow along the pipe, just as water will flow only when there is a

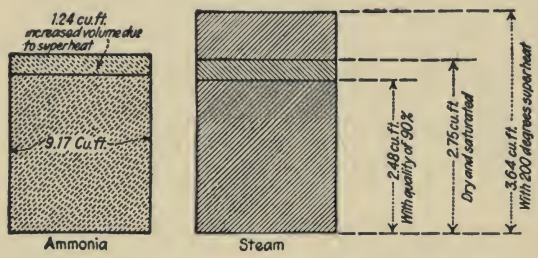

1.24 cu.ft. increased volume due to superheat
9.17 Cu.ft.
Ammonia
Steam
2.48 cu.ft. With quality of 90%
2.75 cu.ft. Dry and saturated
3.64 cu.ft. With 200 degrees superheat

Fig. 89. Figure to represent the effect of quality, or superheat on the volume of one pound of vapor

head to cause the flow. This drop of pressure is increased when the connection between the compressor and the expansion coils is increased in length, when the pipe is decreased in diameter or when fittings, valves, ports and passages offer resistances and cause whirlpool and eddy currents in the gas as it passes along the pipe and passage.

The bad effect of excess condenser pressure, due to resistances to flow of the compressed gas into the condensers, and of wire drawing of the suction gas has been given considerable attention already. The first is reacted directly in the power required to drive the compressor, and the second causes a loss of capacity of the compressor and an increase in the

power required per ton. Pressure drop in the suction line is a serious matter and cannot be tolerated to much extent. Where the pipe connections in the suction line are short this loss is not serious as a rule, but frequently—in packing houses, cold storage and other plants—the return gas has a long way to travel and the velocity must be kept low in order to prevent excessive losses. But before going into the design of ammonia lines, a few other points must be considered.

Figure 89 shows the effect of the quality on the specific volume; that is, the volume of one pound of the substance. For example, the volume of one pound of ammonia at zero degrees F. boiling temperature is 9.11 cu. ft. for dry and saturated ammonia and it is 10.41 cu. ft. for superheated steam with 50 deg. F. superheat. Therefore there will be an increase of 14.3 per cent in volume in being careless in the use of the expansion valve and by allowing the gas to return to the compressor suction bends in a superheated condition. Usually it is desirable to bring the frost back to the suction bends, because this is an outward sign that the gas within the pipe is considerably below 32 deg. F. But if the gas may be reduced in volume by eliminating superheat in the suction headers, why not go a step farther and decrease the volume a little more?

Wet compression, the result of bringing wet ammonia into the cylinder, has few advocates at present. The same liquid ammonia is believed to give much better results by being allowed to perform its proper function of providing refrigeration in the cooling coils. In fact it seems a very roundabout method of cooling the gas on the compression stroke by spraying liquid into the cylinder. Of course the temperature of the gas discharging from the cylinder is reduced, but at the expense of liquid suitable for cooling effect in the expansion coils. The important consideration is which condition will give the greatest amount of *useful refrigeration* per horse power or per kilowatt input. As a rule American engineers

believe that the most economical operation is obtained with the gas returning to the compressor in a dry, saturated condition. Besides this, the use of liquid injection into the cylinders does not lend itself to easy control without constant care. This brings up certain troubles encountered in the expansion coils.

The expansion coils may be designed for feeding at the top or at the bottom. With the expansion valve at the top it is believed that there is less danger from slugs of ammonia returning to the compressor. The reason for this is that, as the liquid cannot collect appreciably in the coils, the feed cannot very well be of the wrong amount without an indication on the suction bends or the compressor cylinder. Liquid return in certain designs of compressors will cause the stuffing box to leak ammonia badly, and in any case there will be heavy frost as far as the machine. In these cases about the only chance of "slugs" of liquid is where the pipe allows pockets from which the liquid ammonia might be picked up by the return gas. Yet, judging from steam engine practice, such slugs of liquid are usual only when a great sudden demand for the vapor takes place, due to a change of load which causes an appreciable increase in the vapor velocity.

On the other hand the liquid feed at the bottom of the coils allows them to be operated when flooded with better heat transference. Being flooded the coils may fill completely with liquid (as is frequent if the suction pressure is not low enough to cause boiling in the expansion coils) and "lost" or "dead" ammonia results. In addition the liquid is likely to *slop* over and cause liquid hammer or worse in the compressor. Where the expansion valve is placed at the bottom of the coils it is wise to use a large liquid separator in the suction line, or a large header arranged in a manner similar to Fig. 95.

Pipe Calculations. In the refrigerating system the following pipe calculations are necessary. *First,* the connections from the compressor to the condenser to convey the super-

heated gas at an average pressure very nearly that of the condenser. This may vary from 100 to 225 lbs. gage depending on the temperature leaving the condenser of the cooling water used, and this in turn is dependent on the initial temperature and the amount of the cooling water. *Second,* the liquid line connecting the liquid receiver with the expansion valve. *Third,* the suction return from the expansion coils to the suction bends on the compressor. *Fourth,* the amount of lineal feet of piping to be used in the ammonia expansion system.

The discharge line should be given careful attention. It is undeniably true that the pressure *in the condenser* is determined by the condensing water temperature and the *effective* area of the condenser, resulting in a certain so-called vapor tension of the condensed (liquid) ammonia, but this is not the pressure against which the compressor is pumping. The compressed gas is made to pass through the ports and valves of the compressor head, then through the discharge bends, past many valves and fittings and perhaps through a long pipe connection (frequently from the basement to the roof) before reaching the condenser and the condensing water. The compressor has the same *practical* condition as exists in the case of a water pump which discharges into a tank situated at some elevated position. The pump has to work against the static head plus the resistances, which vary directly as the length and inversely as the inner diameter and it is increased by the kind and the number of the fittings used. Likewise the steam engineer who desires a steam pressure of 175 lbs. at the throttle notes that whereas the boiler pressure has to be but slightly in excess of 175 lbs. at low loads yet frequently at full loads or at overloads the boiler pressure has to be considerably in excess of 175 lbs.

In like manner the suction pressure is maintained at a certain amount in the refrigerating coils. The pressure has to be a certain amount if refrigeration is to be obtained at a par-

ticular temperature. These temperatures are prescribed for each case by the kind of refrigerating work being performed any may vary from 40 deg. F. to minus 10 deg. F. or lower and in order to provide the required temperature the boiling temperature on the ammonia has to be some 10 or 15 degrees lower still. This suction pressure is maintained by means of the compressor which pumps from the expansion coils per minute as great a weight of ammonia as is evaporated every minute. The effective piston displacement is equal to the weight of ammonia evaporated times the volume of one pound of ammonia at the suction pressure.

But naturally the pressure in the inside of the compressor must be less than that in the suction bends, and these latter have to be less than the pressure in the expansion coils for otherwise the gas would not flow in the proper direction. The case is somewhat similar to the condition of a flowing brook; if the water flows, then there is a difference of level. It requires some difference of level (or pressure) to mainatin a flow of any fluid against the skin or other resistances of the pipe, duct or channel. Therefore in the case of the suction line of the ammonia system there will be a drop of pressure to get the gas into the compressor.

The Diagram. The whole matter of pressure drop may be made clear by referring to the diagram, (Fig. 90). This diagram, for convenience, is worked out for a straight length of pipe of 100 ft. Pressure drop increases with the density of the substance and as the square of the velocity. To get this diagram on the paper to best advantage it has been found advantageous to calculate the lower curves for a uniform density corresponding to 20 lbs. gage and with different gas velocities up to 6,000 feet per minute. For any other density it is necessary only to project upwards until the proper gas pressure is obtained. For example a 4 in. pipe with gas at 60 ft. per second would have a pressure drop of 0.33 lbs. per sq. in. per 100 ft. of straight length for 20 lbs. gage

suction pressure, but this pressure drop would be 1.9 for 200 lbs., 1.47 for 150 lbs., 1.03 for 100 lbs., and only 0.15 for zero pounds gage pressure. Valves and fittings increase the resistance greatly, which vary from 100 times the diameter of

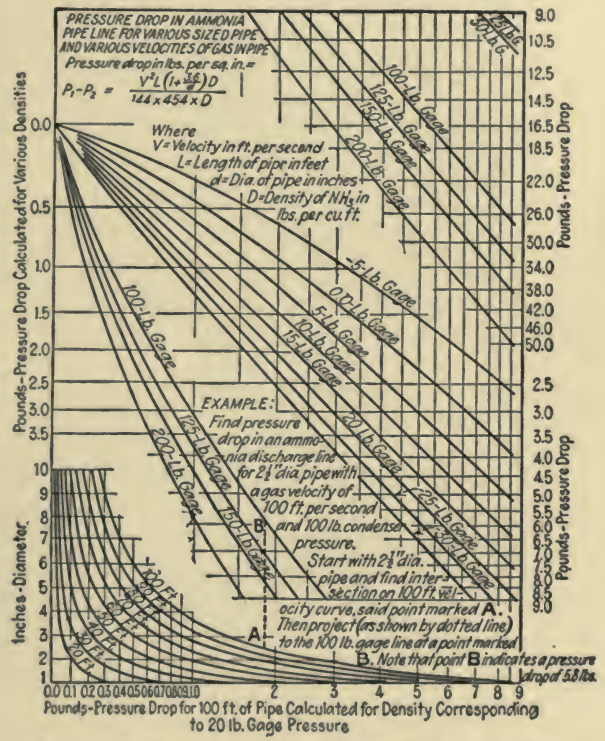

PRESSURE DROP IN AMMONIA PIPE LINE FOR VARIOUS SIZED PIPE AND VARIOUS VELOCITIES OF GAS IN PIPE
Pressure drop in lbs. per sq. in. =
$P_1 - P_2 = \frac{V^2 L (1 + \frac{36}{d}) D}{144 \times 454 \times D}$
Where
V = Velocity in ft. per second
L = Length of pipe in feet
d = Dia. of pipe in inches
D = Density of NH_3 in lbs. per cu. ft.
300 lb. Gage
200 lb. Gage
150 lb. Gage
125 lb. Gage
100 lb. Gage
50 lb. Gage
25 lb. Gage
10 lb. Gage
5 lb. Gage
0 lb. Gage
EXAMPLE:
Find pressure drop in an ammonia discharge line for 2½" dia. pipe with a gas velocity of 100 ft. per second and 100 lb. condenser pressure.
Start with 2½" dia. pipe and find intersection on 100 ft. velocity curve, said point marked A. Then project (as shown by dotted line) to the 100 lb. gage line at a point marked B. Note that point B indicates a pressure drop of 5.8 lbs.
Pounds-Pressure Drop Calculated for Various Densities
Pounds-Pressure Drop
Inches-Diameter
Pounds-Pressure Drop for 100 ft. of Pipe Calculated for Density Corresponding to 20 lb. Gage Pressure

Fig. 90. Diagram showing the pressure drop in suction and discharge ammonia lines

the pipe for a 90 degree ell to 200 times the diameter for a globe valve. Irregularities of the interior, which will cause whirlpool or eddy currents increase the pressure drop. The result is that the designing engineer has to calculate the entire effective length of the pipe, allowing for each ell, tee, coupling,

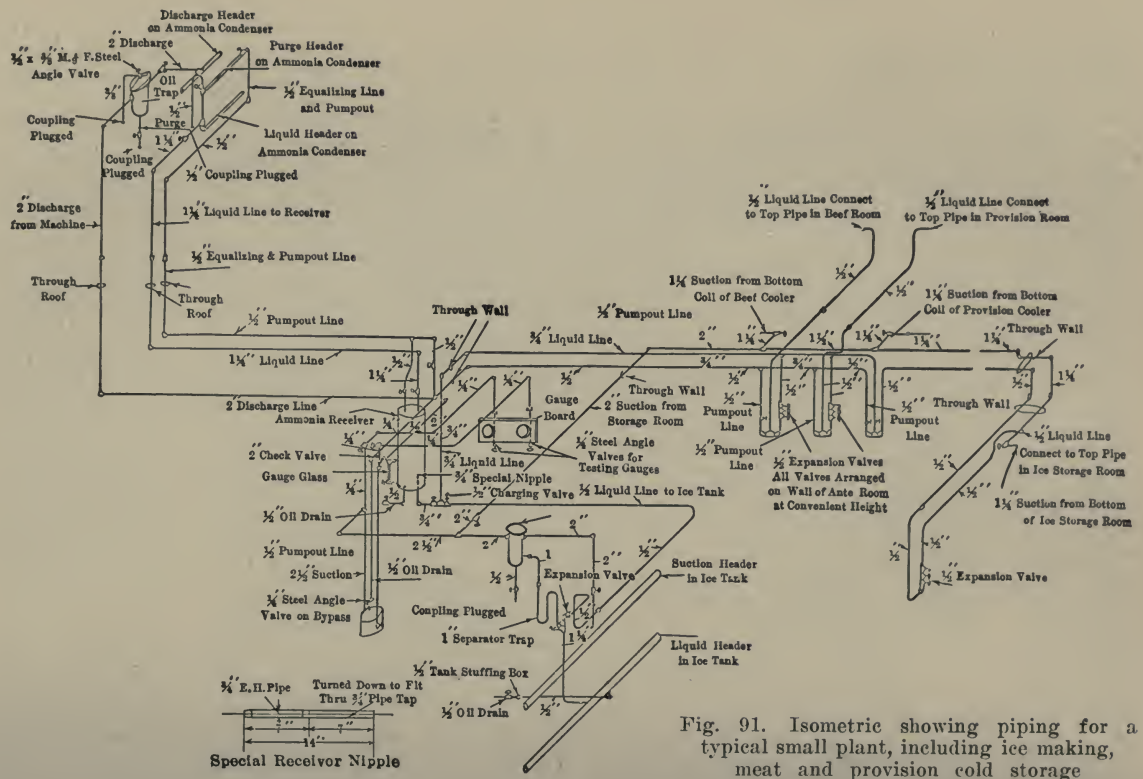

Discharge Header on Ammonia Condenser
2" Discharge
Purge Header on Ammonia Condenser
Angle Valve
Oil Trap
½" Equalizing Line and Pumpout
Coupling Plugged
Purge
Liquid Header on Ammonia Condenser
Coupling Plugged
½" Coupling Plugged
2" Discharge from Machine
1¼" Liquid Line to Receiver
½" Equalizing & Pumpout Line
Through Roof
Through Roof
½" Pumpout Line
Through Wall
1¼" Liquid Line
2" Discharge Line
Ammonia Receiver
2" Check Valve
Gauge Glass
¼" Oil Drain
½" Pumpout Line
2½" Suction
¼" Steel Angle Valve on Bypass
½" Oil Drain
¾" Liquid Line
¾" Special Nipple
½" Charging Valve
Gauge Board
2" Suction from Storage Room
Through Wall
¼" Steel Angle Valves for Testing Gauges
¾" Liquid Line
½" Liquid Line to Ice Tank
Expansion Valve
Suction Header in Ice Tank
Coupling Plugged
1" Separator Trap
Liquid Header in Ice Tank
½" Tank Stuffing Box
¼" E. H. Pipe
Turned Down to Fit Thru ¾" Pipe Tap
½" Oil Drain
Special Receivor Nipple
½" Liquid Line Connect to Top Pipe in Beef Room
½" Liquid Line Connect to Top Pipe in Provision Room
1¼" Suction from Bottom Coil of Beef Cooler
½" Pumpout Line
¾" Liquid Line
1¼" Suction from Bottom Coil of Provision Cooler
Through Wall
Through Wall
½" Pumpout Line
½" Pumpout Line
½" Pumpout Line
½" Expansion Valves All Valves Arranged on Wall of Ante Room at Convenient Height
½" Liquid Line Connect to Top Pipe in Ice Storage Room
1¼" Suction from Bottom of Ice Storage Room
½" Expansion Valve

Fig. 91. Isometric showing piping for a typical small plant, including ice making, meat and provision cold storage

valve or other fitting and find the approximate pressure drop from the diagram. Then, by using a liberal factor of safety, the practical working value may be closely approximated. Figure 91 shows an isometric of a typical small piping layout.

Velocity of Gas. The question may arise as to how the velocity of the gas in the pipe may be found. This is determined by dividing the total volume of the gas passing through the pipe per minute by the cross sectional area of the pipe. The volume is found from the ammonia tables in Chapter VI. For example, a 50 ton machine pumps against 175 lbs. gage and with a temperature of discharge of 200 deg. F. and a suction pressure of 15 lbs. gage. The specific volume at 175 lbs. and 200 deg. F. temperature is 2.03 cubic feet and at 15 lbs. gage it is 9.32 cubic feet for dry, saturated ammonia. At 15 lbs. gage boiling pressure of the ammonia there will be required $\frac{200}{r_{15} + q_{15} - q_{175}} = 0.428$ pounds of ammonia per minute per ton of refrigeration. Naturally the amount required for 50 tons will be 50 times 0.428 or a total of 21.4 pounds, which will result in $21.4 \times 9.32 = 199.5$ cu. ft. of suction gas, and $21.4 \times 2.03 = 43.5$ cu. ft. for the discharge gas per minute. In a practical problem it is not wise to try to figure too closely and therefore the warmest condensing water should be used in determining the condenser pressure, as well as the lowest suction pressure which experience has shown is likely to be used in getting the work out. Dividing the volume in cubic feet by the area of the cross section of the pipe in *square feet* will give the required velocity of the gas, *in feet per minute.*

In the foregoing an attempt has been made to show the factors influencing the design of the suction and the discharge pipes. It is important to note that whereas the first cost of the installation must sometimes be kept down to the practical minimum, yet daily operation must not be handicapped by faulty designs. The refrigerating plant, like any

group of apparatus, must be balanced in every part, and as a rule the part of the plant found to be the cheapest is the piping. By this is meant that the larger the amount of piping used for a certain particular load, the nearer the suction pressure may approach the pressure corresponding to the room or brine temperature. And the higher this suction pressure the lower the cost of refrigeration. One good example of this principle is shown in the next chapter, on ice making.

CHAPTER IX

Ice Making

Before going into the applications of refrigeration and in particular into ice making it is wise to take breath for a survey of the ground already covered, and to see wherein lies the connection of the different refrigerating industries to what has already been said. To begin with, the high pressure side, i. e., the compressor and the condenser, is identical for different applications of refrigeration with the possible exception of the case of low temperature work. Therefore whatever particular characteristics appear in the different cases involving mechanical refrigeration should be in evidence on the low pressure side, in the details of the expansion coils and the uses and arrangement of these coils, or of the air or liquid made cold by the boiling of the liquid ammonia, or other refrigerant. In addition, the refrigerating engineer has to deal with certain other matters associated with the actual production of cold, as for example the softening of water to be frozen, the compression of air for agitation and the pumping of well water or of cooling of condenser water, all matters of importance.

Application of Refrigeration. Yet for all the differences in the various applications of refrigeration, as in ice making, ice cream making and hardening, cooling of water or of air or in the work involved in packing houses and cold storage problems, the underlying principles are the same. The refrigerant boils in the pipes or in a pressure vessel and the

boiling temperature for the pure refrigerant depends on the absolute pressure at which boiling takes place. The condition is similar to what takes place in a steam boiler where heat is absorbed by the water in the boiler from the gases due to the combustion of the fuel, in consequence of which the gases become colder and the water becomes hotter and is vaporized. On vaporization into steam more water is pumped into the coils (the boiler) and the steam leaves the coils for another part of the system. The use of the steam is as varied as is refrigeration, for the steam may be utilized for prime mover work of various sorts, for heating or for industrial cooking and drying. The water on being pumped into the standard boiler does not "flash into steam" when operated according to safe methods and neither does ammonia (or any other refrigerant) "flash" or "expand" into a vapor on being admitted into the expansion coils.

From the foregoing it is evident that the steam engineer is not dealing with a new principle when studying refrigeration. In the steam heating system steam at from 1 to 10 lbs. is admitted to the steam coils or radiators. The steam gives up most of its heat to the surroundings, the air in the room, and in so doing it is condensed. The steam radiators are a sort of condenser. Likewise the refrigerating coil is a boiler, for the liquid is boiled into a gas, and as a gas or a vapor it passes out of the coil and into the compressor again. If the boiling temperature of the ammonia in the refrigerating coils is too high, as when the suction pressure rises too high, the liquid will lie dormant in the coils like water in service pipes. It can boil and refrigerate only when the surrounding brine or air or other fluid is at a greater temperature than the boiling temperature of the ammonia in the coils at the moment.

In this book an attempt has been made to show that the compressor and the high pressure side are only a means of using the same refrigerant continuously, and that the piston

displacement per minute determines the tonnage possible; also that the valves, passages, and headers affect the economy of operation. Likewise the condenser and the condenser water affect the head pressure, which should be as low as possible everything considered, since more *work* is done during compression and the volumetric efficiency is decreased (the capacity of the compressor is lowered) by an increase of compressor pressure. In a similar manner the suction pressure must be kept as high as possible because the work done in the compressor, the volumetric efficiency, and the displacement of the piston per ton of refrigeration are affected adversely by a suction pressure which is lower than that really required.

Hence, there has been a discussion of the compressor, the condenser, stage compression, ammonia fittings, the brine system, refrigerants and condensing water. So far in this book greatest emphasis has been given to the compressor room.

The applications of refrigeration are independent of the form of compressor or condenser and affect only the low pressure side. Even the use of brine does not affect particularly the engine room equipment except in the required use of a brine cooler and the necessary brine pumps. Of course the suction pressure for indirect refrigeration by the use of brine has to be lower than when direct expansion in the expansion coils is used. This is because the ammonia cools the brine, and then the brine cools the air or other goods requiring refrigeration. This makes a double temperature difference, or heat transfer, and it has been shown that for maximum efficiency of operation the suction pressure (and therefore the boiling temperature of the ammonia) should be as high as is practically possible.

And therefore, for the remainder of this book, particular emphasis will be placed on the suction pressure end of the cycle, and the details of the construction for the main subdivisions in the applications of mechanical refrigeration. Of

all these applications at present the most important is *ice making.*

Ice Making. *Ice making* has assumed very large proportions at the present time even in regions of a good natural crop. There are several reasons for this. Occasionally the natural crop is a failure, and an ice shortage is experienced. Lack of ice is a serious matter not only for the household

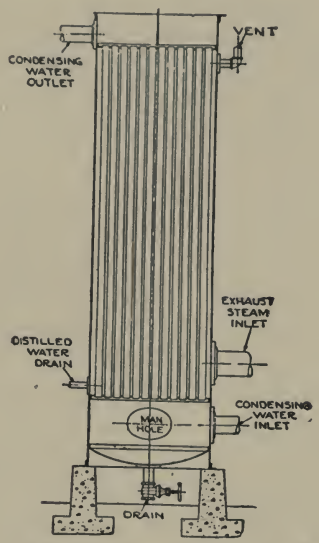


Fig 92. One design of a distilled water apparatus for ice making

but also for those trades requiring icing during shipment or in the process of the industry. But more important at present than anything else is the cost of transportation of the rather bulky and awkward commodity, which shrinks in weight during shipment and which does not lend itself readily to much handling. In fact, the artificial ice, which is made at convenient center points for the trade it is catering to, can compete easily with the natural ice industry in the

matter of the cost of production in all except special locations in the United States. The matter of the purity of the ice need not enter into consideration since artificial ice, even distilled water ice, with a short time interval between making and using is just as likely to be contaminated with bacteria as natural ice with from four to six months storage, except in waters of much pollution.

The ice making industry, by a process of growth and improvement of details, has now but one principal method of manufacture (the can system) but it has several different processes in this system, depending on whether the distilled water, or raw water ice is desired. The so-called *plate* system

TABLE 18. ICE CANS.

Weight of Cake of Ice	Inside Dimensions			Length Over All	Thickness of Material—U. S. Standard Gauge	
	Top	Bottom	Length		Sides	Bottom
50 lbs.	8 x 8	7½x 7½	31	32	No. 16	No. 16
100 lbs.	8 x16	7¼x15¼	31	32	No. 16	No. 16
200 lbs.	11½x22½	10½x21½	31	32	No. 16	No. 16
300 lbs.	11½x22½	10½x21½	44	45	No. 16	No. 16
400 lbs.	11½x22½	10½x21½	57	58	No. 14	No. 14

The above sizes are in accordance with the standard adopted by the Ice Machine Builders Association of the United States. Cans are made throughout of galvanized material, well riveted and soldered and guaranteed tight. Cans made of No. 16 gage material will be turned over top and bottom. The 200, 300 and 400-pound cans have ¼ x 2-inch galvanized bands at top. Small sizes have ¼x1½ inch bands; ⅝-inch lifting holes are punched through bands.

is still used, but it is not being installed in new plants to much extent.

The Plate System. At first ice was made by freezing on one side of a flat plate immersed in a tank of water until 8 to 12 inches thickness of ice was formed, the plate being arranged so as to allow the liquid ammonia to be fed into it suitably and the gas drawn out again to be returned to the compressor in the usual manner. This method of ice making is a raw water system, and such an ice plant is highly successful, and is still used with great satisfaction. However, the plate ice system is seldom installed now because of the

greater first cost required, the greater head room necessary, and the lower back pressure required of the machine. Large cakes of ice are frozen—from 3 to 5 tons—and this requires heavy handling apparatus, a suitable saw and table for sawing into convenient (300 lb.) cakes. The time of freezing is increased greatly with the thickness of the ice made and frequently the time necessary to freeze a cake is six days. However, the product of the plate system is an excellent raw water ice. It has a very clear appearance, but is irregular in size—both as regards thickness and the ability of the ice to pack well in storage. In general, it may be said that all water has air in solution, the amount depending on the temperature of the water and other conditions. If this water is frozen quietly the air in solution will be pocketed in the ice and the ice formed will have the appearance of marble, and is called "tombstone" ice. It may be (and usually is) as valuable as clear ice as far as its cooling effect is concerned, but it does not look as well. The household trade does not care for it and will pay a greater price for the crystal ice. In consequence, water to be frozen into plate ice (and can ice as well) is agitated by means of compressed air under a few pounds pressure which is allowed to bubble up along the freezing surface of the ice and thereby prevent the air of solution adhering to the ice surface.

The Can Ice. Parallel with the development of plate ice systems came the *can ice* method, using distilled water. If steam is condensed (Fig. 92) and reboiled slightly while exposed to the atmosphere, then cooled without being exposed to the air, it may be frozen into a cake without a core, and it becomes as clear as crystal. In the early days of artificial ice making, and to a certain extent at present, especially where the raw water is heavy in salts, the use of distilled water for ice making has been popular. Being distilled and reboiled it was considered hygienic and therefore appealed to the small retail trade.

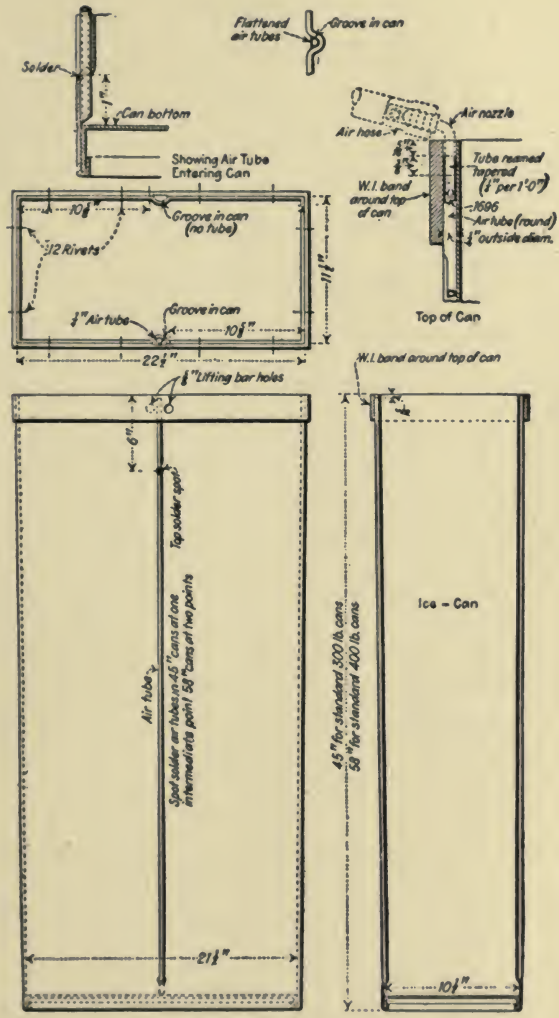


Fig. 93. The high pressure system of can ice agitation

In the can system a galvanized iron can of rectangular cross section (Table 18) but with a slight taper towards the bottom (Fig. 93) is filled with water and is lowered into a brine tank. The brine tank, usually a sodium chloride brine, is maintained at about 14 deg. F. by means of ammonia expansion piping arranged inside the tank on two sides of the cans, or is cooled in a brine cooled either inside or outside the tank. The brine is circulated by means of propellers, with either horizontal or vertical shafts. These propellers are designed in a similiar manner to a ship's propeller and are used to keep a rapid circulation of the brine in and about the cans, thereby maintaining a uniform temperature of brine and also increasing the coefficient of heat transmission through the can to the ice and the water. Heat is abstracted from the water, and the ice is formed rapidly at first, freezing at the start all around the sides of the cans and then at a progressively slower rate as the center is approached. Evidently, with good agitation, the time of freezing of a certain sized can depends on the brine temperature and the cross section of the can. The brine temperature depends on the boiling temperature of the ammonia (the suction pressure) and the linear feet of expansion piping in the tank. All these variables are given in the diagram, (Fig. 94.) This diagram is recommended for conservative practice, and the tonnage of a plant designed by this diagram should be exceeded greatly if care is observed in the operation and, as in Figs. 95 and 97, provision is made so that the amount of agitation can be increased. The flooded system works well in the ice plant, and almost the only care necessary is in the design of the piping in order to keep a good circulation of the ammonia, and also so that the gas returning to the machine shall not be too wet.

The Flooded System. As mentioned, the flooded system (Figs. 95 and 96) in ice making has been successful. The essential feature in the system is a device to precool the liquid ammonia being received from the condensers to a temperature

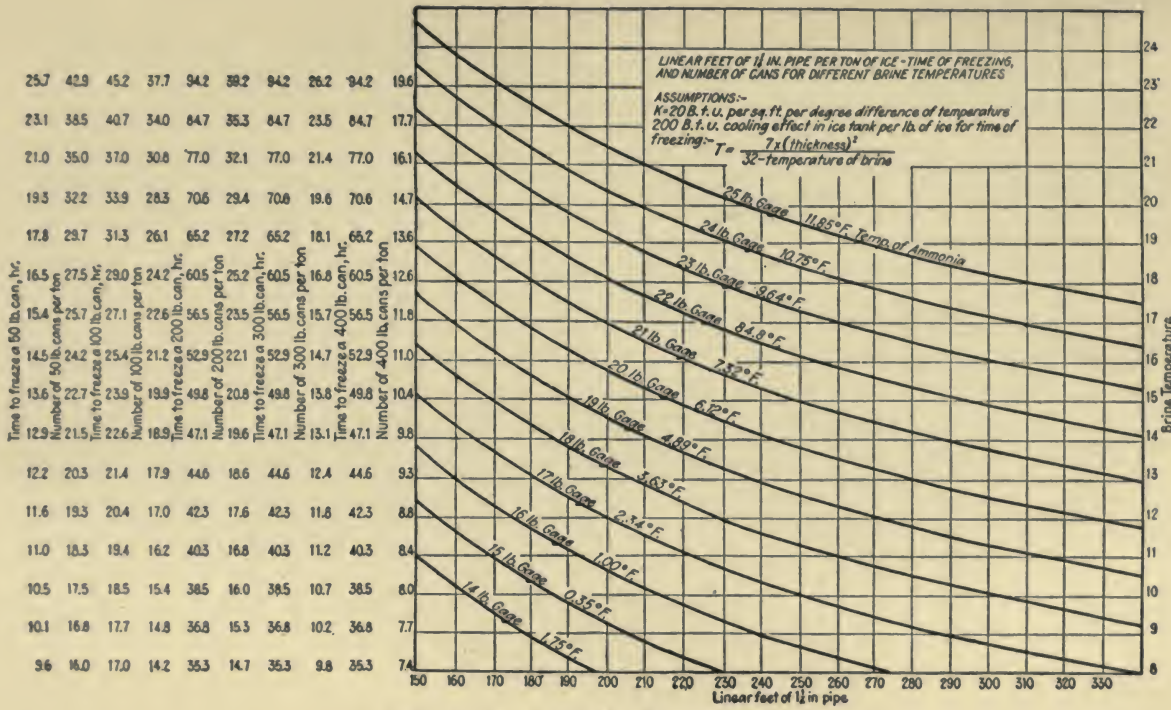

LINEAR FEET OF 1¼ IN. PIPE PER TON OF ICE - TIME OF FREEZING, AND NUMBER OF CANS FOR DIFFERENT BRINE TEMPERATURES
ASSUMPTIONS:-
K=20 B.t.u. per sq.ft. per degree difference of temperature
200 B.t.u. cooling effect in ice tank per lb. of ice for time of freezing:- T = 7x(thickness)² / 32-temperature of brine
25 lb. Gage 11.85°F. Temp. of Ammonia
24 lb. Gage 10.75°F.
23 lb. Gage 9.64°F.
22 lb. Gage 8.48°F.
21 lb. Gage 7.32°F.
20 lb. Gage 6.12°F.
19 lb. Gage 4.89°F.
18 lb. Gage 3.63°F.
17 lb. Gage 2.34°F.
16 lb. Gage 1.00°F.
15 lb. Gage 0.35°F.
14 lb. Gage 1.75°F.
Linear feet of 1¼ in pipe
Brine Temperature
Time to freeze a 50 lb. can, hr.
Number of 50 lb. cans per ton
Time to freeze a 100 lb. can, hr.
Number of 100 lb. cans per ton
Time to freeze a 200 lb. can, hr.
Number of 200 lb. cans per ton
Time to freeze a 300 lb. can, hr.
Number of 300 lb. cans per ton
Time to freeze a 400 lb. can, hr.
Number of 400 lb. cans per ton

Fig. 94. Diagram for can ice making

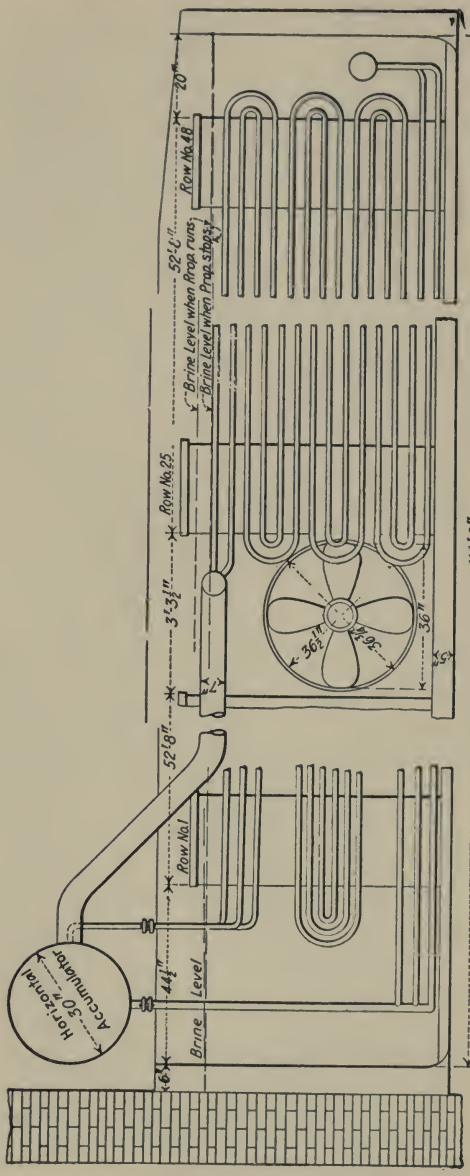


Fig. 95. The flooded system of can ice making. [Note large brine agitation and great difference in brine level]

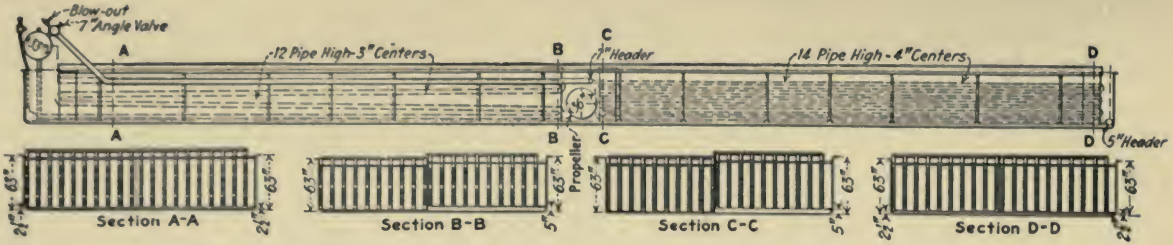


Fig. 95A. A special arrangement of cans to operate with high velocity brine agitation

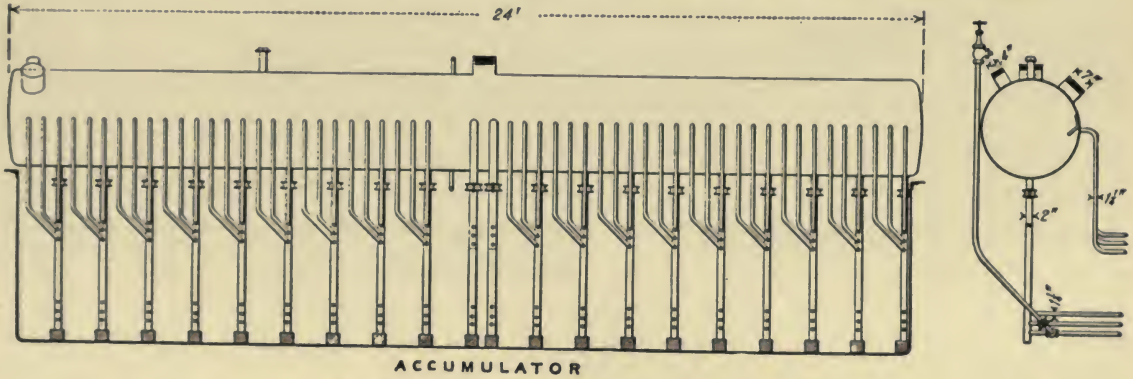


Fig. 95B. The accumulator in Fig. 95

equal to that of the ammonia in the cooling coils (corresponding to the suction pressure). The coils may be fed by gravity or by the usual expansion valve but arranged in a special manner similar to Fig. 96. In either case the coils are much more nearly filled with liquid ammonia and the heat

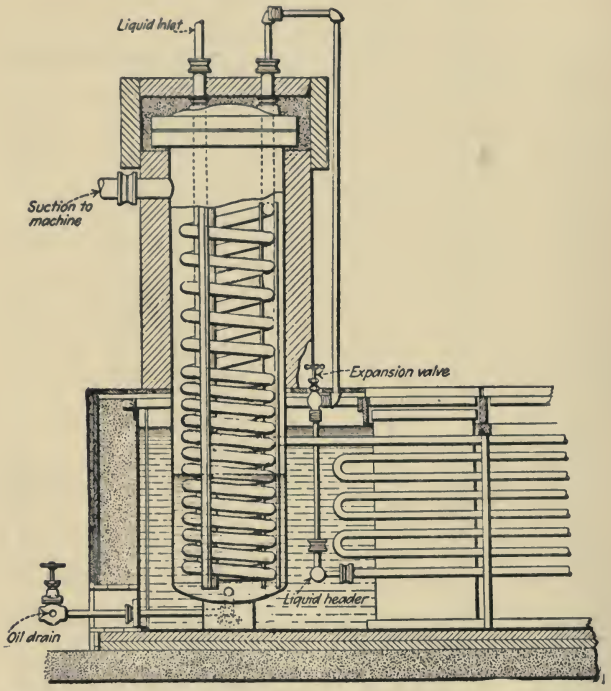


Fig. 96. The York type of flooded can ice making plant

transfer from the brine to the ammonia is better than would be the case with the old system. The flooded system does *not* give *more refrigeration* per pound of ammonia; it simply reduces piping in the tank.

Distilled Water System. The distilled water system is giving way to the raw water ice in regions convenient for

electric power and where the water to be used in the cans is very good; but where the water contains certain solids, or where steam power is necessary it is probable that it will continue to be used. However, unless a distilling system is used, the capacity of the plant is limited to the amount of steam used in the engine or the steam auxiliaries, and this usually means inefficient operation in order to get enough condensed steam for ice making. If fuel is expensive or boiler troubles are bad on account of poor boiler feed water, the use of distilled water is subject to considerable doubt unless it can be shown that the water cannot be softened by any of the known methods.

Refrigeration Required. In can ice manufacture it is clear that two distinct steps are used; the cooling of the water from 70 deg. F. or more often as a rule to 32 deg. F. and then freezing the water and finally cooling the ice to or near the temperature of the brine. If water enters the plant at 70 deg. F. and brine is carried at 14 deg. F., then there is required:

$$\underset{\text{to cool the water.}}{(70 - 32)} + \underset{\text{to freeze the water.}}{144} + \underset{\text{to cool the ice.}}{\frac{32 - 14}{2}} = 191 \text{ B.t.u. of refrigeration.}$$

And the amount of cooling of the water to reduce the temperature to 32 deg. F. is $\frac{38}{191}$ or 20 per cent. This cooling can be done at a higher temperature of the ammonia than that of the freezing process, say, at 40 lbs. gage as compared with 20 lbs. gage for the final freezing, if the parts of the process can be separated. This problem is similar to that encountered in the use of the steam economizer, in making steam, where the boiler feed can be *heated* very advantageously *outside* the boiler by the use of heating surfaces in the path of the flue gases. The advantage of using economizers lies in increasing the temperature of the substance

inside the piping. In cooling water for can ice advantage is obtained in doing the job *outside* of the cans because of the fact that the cooling can be done better (with less surface and quicker) than by the can method, and because the boiling temperature of the ammonia at 40 lbs. gage is much more economical of power supplied than if all the refrigeration were done under one suction pressure. In general practice either a small compressor would carry the precooling load (the usual manner) or the compressor could be arranged as described in Chapter II for compression with two suction pressures.

The Raw Water Can Ice. One of the great advances in refrigeration during the last decade has been in the methods of ice making. The universal use of electric power, generated by means of steam and by hydro-electric plants, has created a demand for an ice plant which does not require the use of a steam generating plant for distilled water. Also the mounting cost of fuel, notwithstanding the reduced steam consumption of the poppet valve and the uniflow steam engine, as well as the application of Diesel and semi-Diesel engines for compressor drives, has resulted in a great demand for a raw water can ice system which can provide a clear crystal like cake of ice.

As already mentioned, agitation of raw water during freezing will provide a crystal plate ice, and the same is true to a certain extent in the case of can ice. However, a residue is left in the can (a core) which is likely to contain all of the impurities of the water in the can. Very good water (free from suspended and dissolved solids) need be agitated only by air under pressure until complete freezing has nearly been attained, but water containing much solid matter not removed by previous treatment must have a core pump. This pump will remove the remaining small part left from the original canful of water. The resulting cavity may then be

refilled with distilled water, although additional raw water is usually sufficient.

Agitation. The methods of providing agitation and securing clear ice are numerous. One way is to provide drop pipes into the center of the can and to allow air under 2 to 3 lbs. pressure to bubble up. This is one of the simplest and cheapest first cost constructions, but it requires constant care

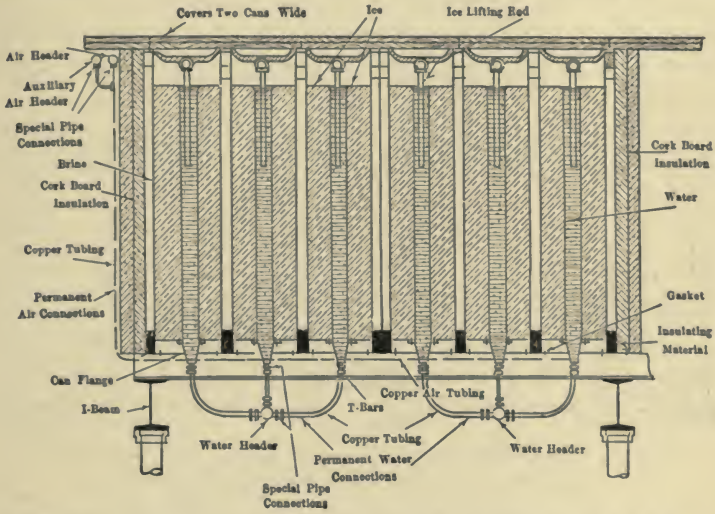


Fig. 97. Cross section through Arctic Fownall ice tank, showing ice about 80% frozen, connections, etc. [Note additional height required to give access to control valves and that steel supports are required for brine tank and cans]

on the part of the operator as the drop pipe may freeze into the ice if it is not removed before the cake is entirely frozen. To prevent this annoyance the high pressure system is used, the air being compressed to 25 lbs. or more and then precooled, either by expanding through a nozzle or otherwise or by means of refrigerating coils, before being allowed to get to the can so as to remove as much of the moisture content as possible. Fig. 93 shows one arrangement where the air

tube passes through a groove in the can, the pipe becoming an integral part of the can, and the air passes up the can as long as there remains a core which will allow air passage. The main objection to this arrangement is the difficulty of removing from the air a sufficient quantity of the water content to prevent the formation of frost at the needle valve and in the feed pipe, thus choking up the air supply to the can and thereby preventing agitation. Also it is necessary to have 100 per cent continuous operation, as otherwise the feed valve will freeze solid and (without agitation) will form opaque ice.

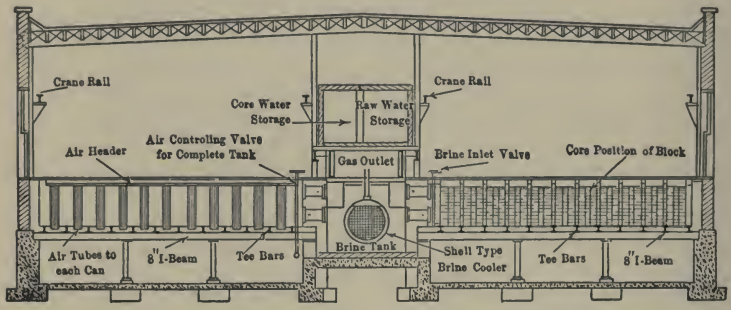


Fig 97A. Cross section through tank room. [Note compact arrangement, but additional elevation required]

The Beal System. The high pressure (Beal) air agitation system then requires considerably higher first cost and more power input per ton of ice—perhaps as much as 16 K. W. hours per ton—over that required by the drop pipe systems using only 2 to 3 lbs. air pressure. The drop pipe system, however, requires more labor to operate the air connections.

The modern tendency is to reduce the amount of labor in ice making as much as possible. This is shown by the applications of special designs such as the high pressure air system already described which reduces the labor of attending to the drop pipes of the low pressure system, and in attempts

like "gang" hoisting can cranes and the use of special tank designs like the Arctic-Pownall ice plant. The latter, shown in Fig. 97, has stationary cans, arranged in tanks of 28 to

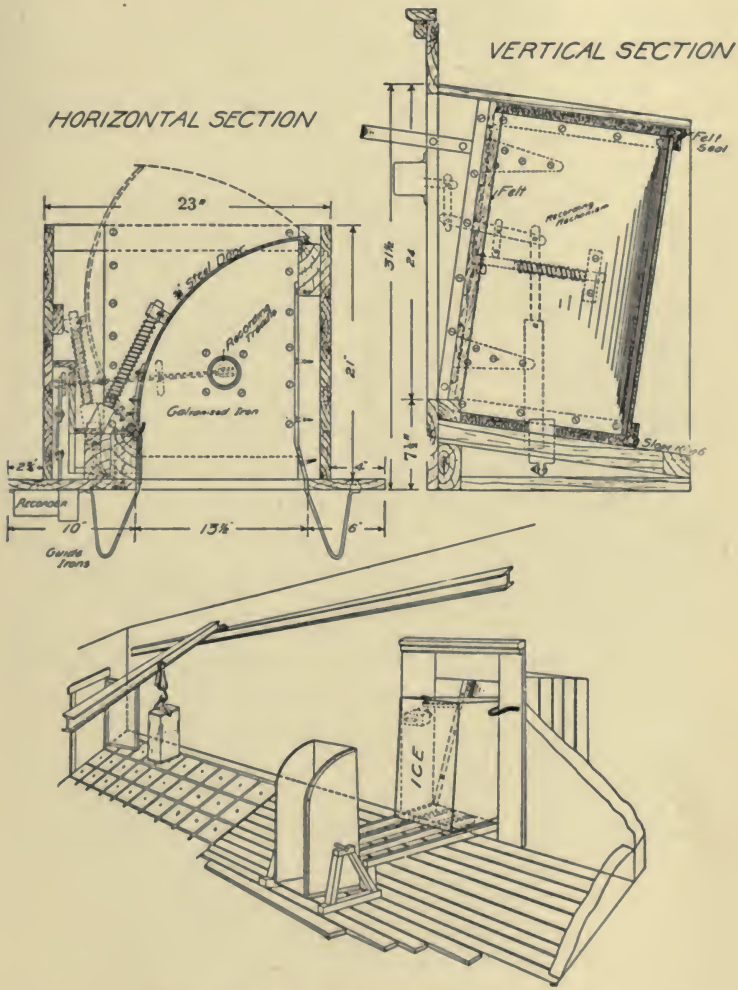


Fig. 98. Detail of ice chutes and doors

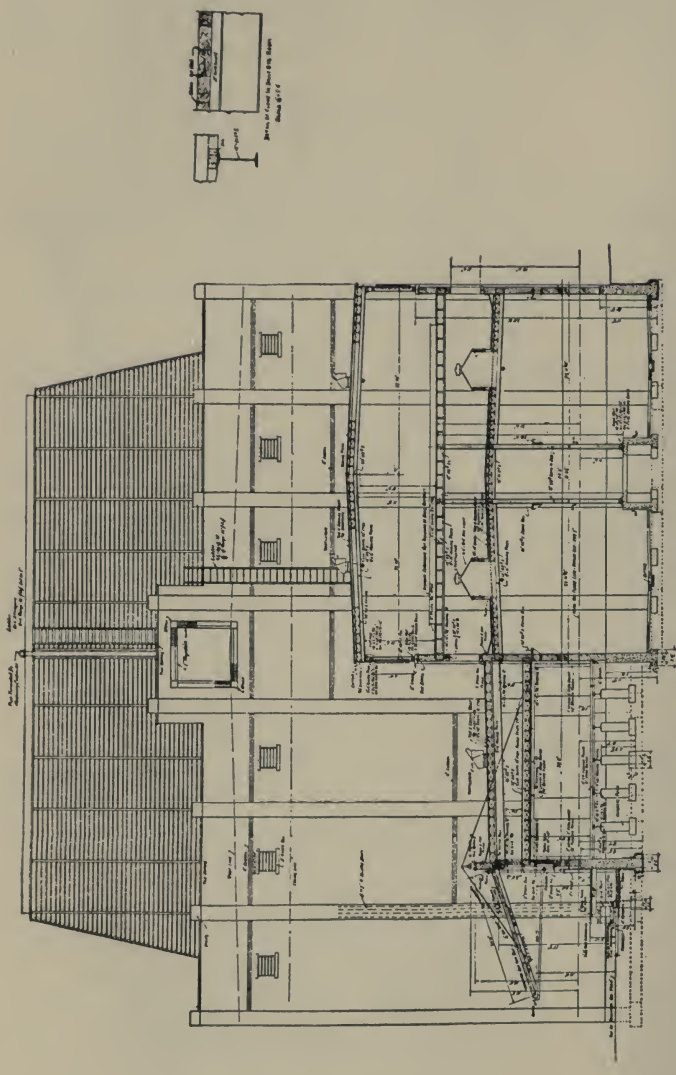


Fig. 99. A modern, typical ice making plant

78 cans each, and devised so that the bottom of the cans is insulated and the filling of the cans, dropping of the cores and air agitation may be controlled by means of one valve each. In thawing, the brine in the tank is shut off from the other tanks and is pumped through the water storage tanks, thereby precooling the water, and (as the brine is heated) thawing the ice in the cans. In filling the cans, overflow valves prevent too much water entering any one of the cans. Evidently the gain is in the control of sets of cans by means of one valve, and in the easy precooling of the water. Quick cooling (as in many new ice plants) is obtained by means of good agitation. The shell and tube brine cooler is used exclusively. Table 19 gives the approximate amount of power required.

TABLE 19.

No. 300-Lb. Cans per Ton Condenser Pressure.	Horsepower Input to Motor, Belted Compressor, with Varying Condenser Pressures.				Kilowatt-Hours per Ton Ice, for Compressor Only, with Varying Condenser Pressures.			
	145 lb.	165 lb.	185 lb.	205 lb.	145 lb.	165 lb.	185 lb.	205 lb.
14	2.08	2.38	2.59	2.84	38.8	43.8	48.3	52.9
15	2.01	2.28	2.56	2.78	37.5	42.6	47.8	51.8
16	1.96	2.21	2.48	2.74	36.5	41.2	46.3	51.1
17	1.94	2.18	2.43	2.68	36.2	40.6	45.3	49.9

In all cases it is assumed that water at a temperature of 70 deg. F. will enter the storage tank for freezing purposes.

The labor due to simply pulling the ice, thawing the ice out of the cans (Fig. 98), returning the empty cans and refilling is a large part of the final cost in producing ice. The use of cranes lifting as many as 12 cans at once is sometimes made, and some plants have shown a decided economy by their use, especially in long tank rooms which require a long carry to the thawing and dumping platform. Fig. 99 and 100 show typical modern ice making and storage plants.

Water Softening. As already mentioned not all water is equally good for raw water ice. Water containing certain solids will cause opaque cores, discolored patches, or even

cloudiness on the outside of the cakes. Much of this trouble may be decreased, or possibly removed entirely by water *softening* or by proper agitation and core sucking.

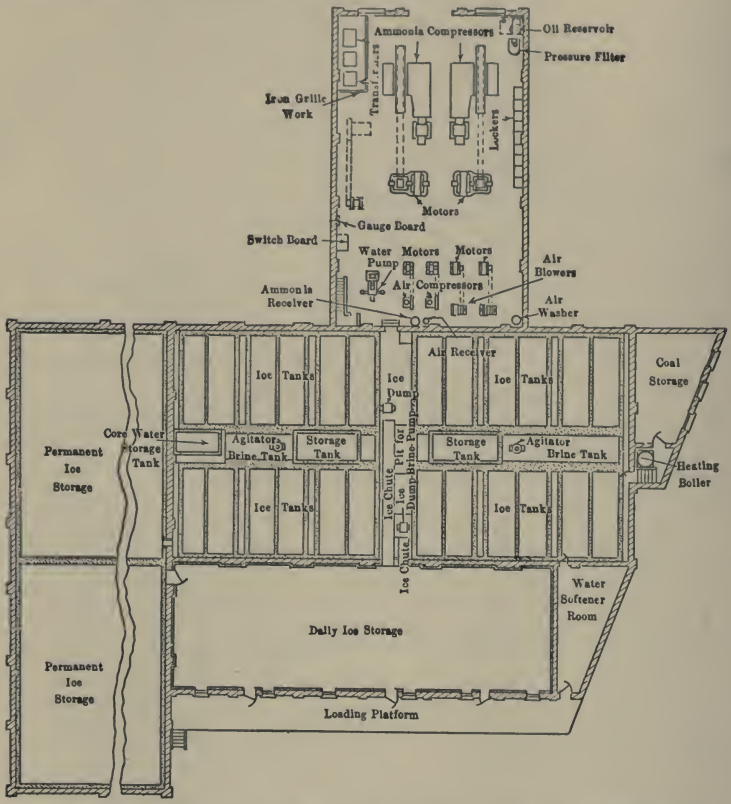


Fig. 100. Floor plan—raw water ice plant, City Ice Delivery Co. Cleveland, Ohio. [Note short run to ice dump, arrangement for ice chutes, and location of daily and permanent ice storage]

Water softening is done in two principal ways:—by the use of lime and soda ash and by the use of zeolite. The latter uses the principle of "exchange" where the carbonates and

sulphates of calcium and magnesium become sodium salts. The total *weight of solids* remains about the same and the only advantage is in the softer water obtainable.

The *second method* of softening using *lime and soda ash* (Fig. 101) requires considerable care, but will actually eliminate the bi-carbonates and will greatly reduce the amount of

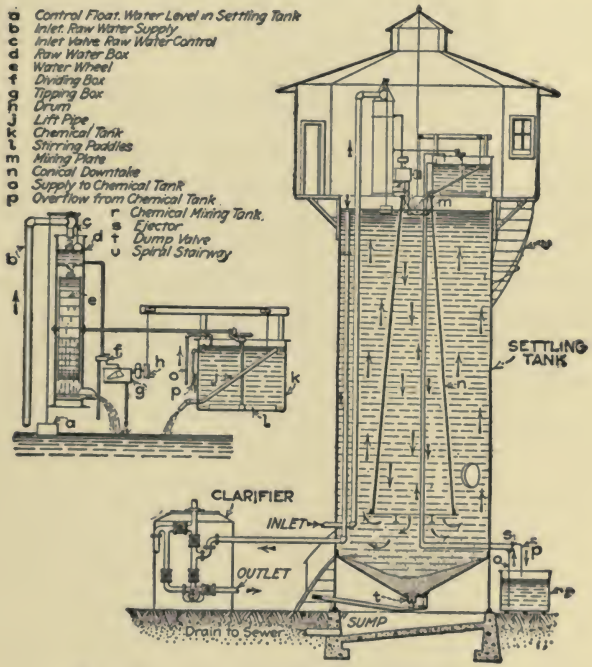


Fig. 101. One system of water purification

carbonates and sulphates present. It will also reduce the tendency towards shattering of the ice, which to a large extent is caused by the presence of solids and carbonic acid, in the water. Other causes of shattering are too cold brine, refilling the core with warm water, poor brine agitation and rough handling of the can during thawing and dipping.

As suggested, the amount of agitation has an effect on the appearance of raw water ice. For example, a 300-lb. can with about four grains of total solids per gallon will require about 0.65 cubic feet of air per minute, but this allowance must be increased with solids in excess of four grains, and an additional allowance must be made for brine temperatures lower than 14 deg. F. Finally it may be said that water high in sulphates cannot be used for raw water ice making, and the attempt should not be made.

There are several other systems of ice making, possessing more or less merit. The main points in raw water ice manufacture are the same for all of the methods, however. "Water good enough to drink is good enough to make ice," and clear raw water ice must be agitated during freezing.

In conclusion it may be said that of all methods the raw water can ice manufacture offers the greatest possibilities for continued progress. Future endeavor is towards the securing of high suction pressures for the ammonia, high values of "k" due to rapid circulation of brine, and in reduced labor costs in handling the cans and the ice.

CHAPTER X

Applications of Refrigeration

Cold Storage, Air and Liquid Cooling

In the preceding chapter the matter of the applications of refrigeration was taken up, and it was shown that the high pressure side was the same in nearly every case, except where stage compression was found desirable or the absorption machine was used. In other words, the application of refrigeration affects the low pressure side, and it is here that the details become varied and numerous. Ice making was described in some detail and the fact that can ice making using raw water in the cans is becoming the favorite method was mentioned and reasons given for the preference. Because of its bulk, ice must not be handled much and is best produced in proximity to the zone where it is to be consumed; for this reason artificial ice is rapidly replacing natural ice even where the latter is plentiful. Next to the importance of ice making comes the business of cold storage, which includes chilling, sharp freezers, and coolers for the preservation and curing of meats, fish, vegetables and fruits. However, before going into the subject of cold storage, it will be wise to give the reasons for its use.

Object of Cold Storage. Meats are chilled quickly because otherwise the natural process of decay will set in. Vegetables, fruits, dairy products, etc., are kept at or about the freezing point of water because it has been found that

the bacilli which cause decay are prevented from growing actively when held at this temperature. Cold storage cannot preserve goods indefinitely (except certain meats when kept at extremely low temperatures) because the bacteria keep growing and increasing in numbers when at cold storage temperatures. A large number of goods cannot be frozen—fruits and vegetables for instance—without spoiling the cell formation and the internal structure of the goods, and therefore such goods cannot be kept in cold storage for more than a few months, if that long.

Humidity. Besides the matter of the temperature, there is the matter of the humidity of the air, which affects not only the holding power of foodstuffs but the quality of these goods as well. This is more marked in some cases than in others, but with eggs too great humidity will cause spoiling and too low a humidity will cause shrinking. In the packing house chill room dense clouds of steam come off from the freshly killed animals, and the manner of refrigeration must also provide a convenient means of removing this vapor. Vegetables, fruits, etc., dry out during the holding process, thus increasing the humidity. As moist air is bad for cold storage goods the refrigerating plant must provide the necessary means for removing this surplus water vapor and keeping the temperature and the humidity correct for the particular goods being stored, these requirements being different for each variety of foodstuffs. The humidity, roughly speaking, must be in the neighborhood of 70 per cent. The accumulation of frost on the expansion piping is an indication of the removal of moisture from the air.

Kind of Piping for Cold Storage. There is no set rule in regard to the type of direct expansion or brine storage piping. The same plant may have both systems, as is frequently observed. Packing houses use brine for the greater part of their work, although one large company, at least, has little or no brine refrigeration. This is because some of the load

is heavy and of short duration, and can be handled well by the use of brine storage tanks and a smaller compressor operating the greater part of the time. Also it is *safer* to use brine where the pipes are exposed and are subject to accident. In addition the chill rooms are usually designed to have a loft with brine sprays or a brine curtain, as brine can pull down the temperature more quickly and absorb the moisture better than any other system. (See Figs. 102 and 103.)

On the other hand, it is possible to operate compressors on the direct expansion system more economically than with

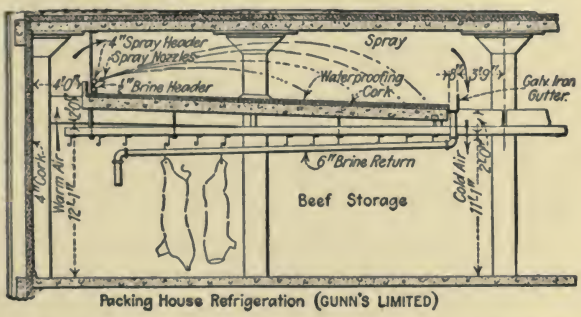


Fig. 102. Brine spray [packing house] refrigeration

the brine system, because the boiling refrigerant cools the air directly instead of indirectly as is the case with the use of brine. But it may be remembered that the direct expansion must be carefully laid out so as to return the gaseous ammonia with the least possible pressure drop to the compressor. In this connection one occasionally sees in the technical press descriptions of changes made in old plants—as, for example, the redesign of the Anheuser-Busch plant in St. Louis—in order to secure more economical operation. The trouble in the St. Louis plant was that it grew like "Topsy" and soon was a mass of large and small pipes of every description. Other factors that must be considered in deciding on the type of

piping are the effect on the goods held in storage of a break in the pipe line, the cost of the ammonia piping as compared with brine piping, cost of the initial charge of ammonia, and its losses due to leaks or corrosion of the pipe line. The brine system is *self-contained,* having, with the exception possibly of the ammonia condensers, the entire high and low pressure side under the easy control of the operator without his having to leave the compressor room. It is to be remembered, however, that extra equipment is necessary with the use of brine, namely, the brine coolers, the brine pumps and the power necessary to circulate the brine. It may be added here again

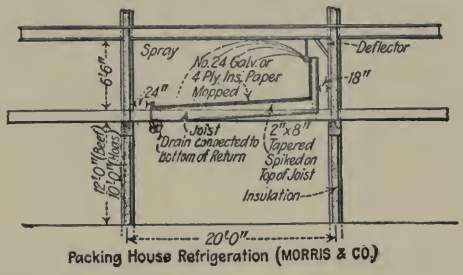


Fig. 103. Brine spray [packing house] refrigeration

that the absorption machine must always use brine, as direct expansion is not so advantageous as in the case of the compression system and the brine system works out most satisfactorily.

The Bunker Coil. To circulate the air better, giving more ready control to both temperature and humidity the outside bunker coil is sometimes used. In brief, this is an arrangement of coils in a separate loft or room, so arranged that the air may be forced through at a moderate velocity (where it is cooled to the proper temperature) and then through ducts to the distributing openings. The return is usually the reverse of the feed lines.

Such a system is sometimes recommended because it removes all piping from the cold storage room. With care in the design a uniform temperature is possible, and there is less danger of excessive cooling in certain parts of the storage room. Power must be provided to drive the fan which must be carefully selected so as to require the least power input for the volume of air circulated. But whereas less piping may be required than with the ordinary system on account of the better heat transmission, there are some objectionable features.

The motive power necessary for the fan also heats the air and requires additional refrigeration to neutralize it. The main objection to bunker coils, however, is in the matter of their cleaning. Usually the coils are close together and combined with baffles, make a difficult piece of equipment to clean. As the ice accumulates on the pipes, the heat transfer rate will drop until either the pipes are cleaned or service is unsatisfactory. This point is mentioned here because in the case of ice cream hardening rooms the bunker coil is being replaced by pipes in the room. The lack of success with hardening rooms is purely a matter of operation, but this service is the "court" of last resort when it comes to the proper design of a new refrigerating room or plant.

The Sharp Freezer. The sharp freezer is the term used for a cold storage room which is held at or near zero degrees F. It may have better insulation than the ordinary cold storage room, usually about six or eight inches of corkboard, and it generally has an ante-room. The piping has to be heavy to carry the heat leakage, which may be as great as double that of the general storage room for fruits and vegetables which are held at from 36 to 40 deg. F. Sharp freezers are designed usually for fish, meats, eggs in bulk, etc. The temperature carried is usually from 0 to 15 deg. F.

Cold Storage Details. It is hardly within the scope of an elementary book to go into detail except as to fundamentals. The storing of goods, the duration of storage, the

height of boxes or packages, the amount of aisle, etc., are important details, but not necessarily of the kind that the operating engineer has to keep in mind. A few figures are included, however, for handy reference. Particular attention is called to the layouts, which are all recent designs and are typical of the kind of service being performed (see Figs. 107 and 108). Also Figs. 102 to 106 give an idea of the construction of piping and cold storage systems in general.

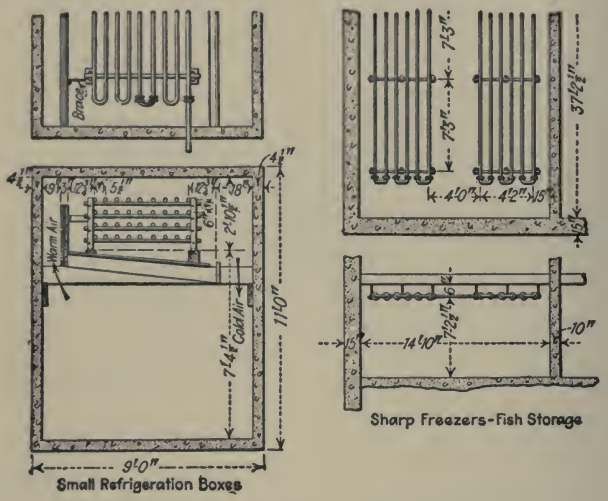


Fig. 104. Typical piping Installations

Do not put second or third grade goods in storage. Cold storage goods should be of the best quality. Fruits and vegetables should be carefully handled to keep them from unnecessary injury, and they must be re-sorted—after a period of time depending on the kind of goods—to remove the sprouts, spoiled parts or spoiled containers.

Fig. 107 and 108 show the ratio of room space to piping and also the room space per ton of refrigeration. In each figure it will be noticed that this ratio is low for small cold

Fig. 105. Typical Brine piping

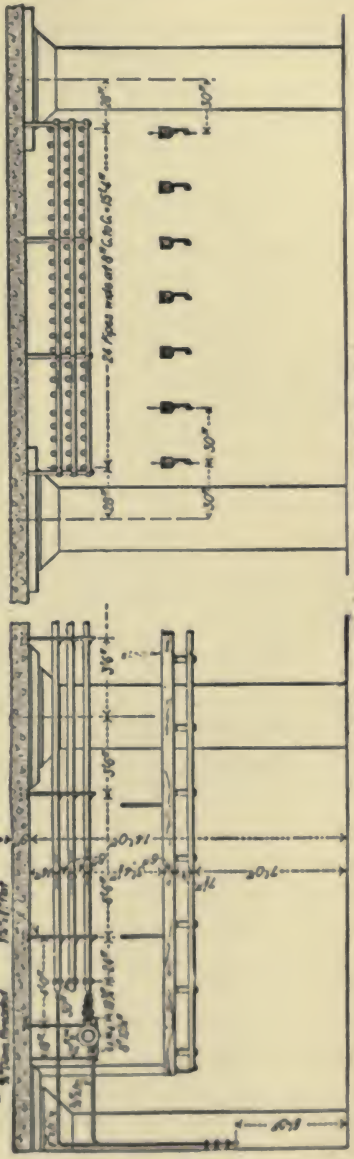


Fig. 106. Sharp freezer (low temperature) piping

storage rooms, increasing in value for large rooms. For rooms that have to be opened frequently the ratio of the piping per cubic foot is greater than for boxes seldom opened, because the entrances of rooms that are to be opened frequently are bound to create losses through the doors. A large saving of refrigeration can be made by the installation of proper door fittings and by all arrangements that help to reduce the duration of opening to a minimum. There is much carelessness in this matter in the practice of many plants.

TABLE 20. COLD STORAGE PRACTICE.

Temperature of Storage, Deg. F.	Humidity, per cent	Allowances
		1 bbl. = 31½ gal. of 231 cu. in. each.
		1 bushel = 1.245 cu. ft. = 2,150.4 cu. in.
35	80 to 90	1 bbl. potatoes = 5 cu. ft. = 180 lb. = 2½ bu.
32 to 34	70 to 80	1 box cheese = 60 lb. = 2 cu. ft.
30 to 31	75	1 case eggs = 30 doz. = 50 — 70 lb. = 2¼ cu. ft. = 12x13x25.
32 or lower	70	1 tub butter = 63 lb. = 1½ or 2 cu. ft. (piles not over 6 ft.)
30 to 35	85	1 bbl. apples = 150 lb., 5 cu. ft. = 2½ bushels.
34	80 to 90	1 crate celery = 10 cu. ft. = 140 lb. = 24x24x30 in. (8 ft. high).
34	80 to 90	1 bbl. vegetables = 5 cu. ft., piles 5 ft. high.
35	80	1 box oranges (packed on end) = 4 cu. ft., 70 lb. = boxes 15x15x30.

Aisles—Allow 1-3 for eggs, celery and oranges.
Allow 1-5 for potatoes and vegetables.
Allow 1-4 for mutton, pork, poultry (4 ft. high).
Specific gravity of boned beef = 1.073 = 67.2 lb. per cu. ft.

Applications of Refrigeration. Some thirty years ago mechanical refrigeration was used mainly in the brewery industry, but of late years the great advance, first in ice making and then in cold storage (including the packing house industry) has relegated the brewery total tonnage—even before the advent of adverse legislation—to secondary importance. And there is now hardly any branch of engineering that is developing with the speed of refrigeration in variety of application, and in the amount of sales of new machinery. The chemical industries are paying more and

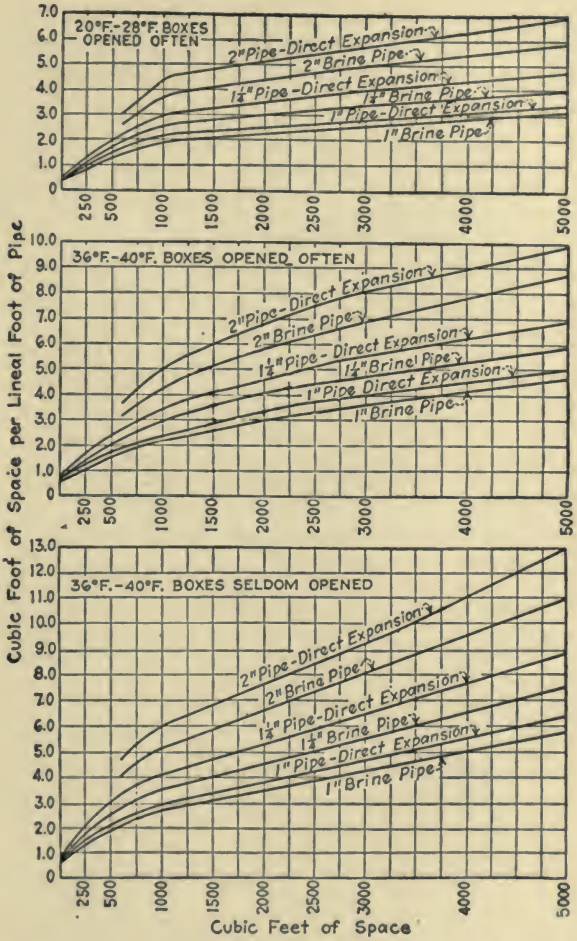


Fig. 107. Cubic feet of space per linear foot of refrigeration piping per ton of refrigeration

more heed to the latent possibilities of the mechanical cooling in their processes, as are the engineers in almost every branch of work. The steel mill uses refrigeration in tempering and hardening, the oil refinery in separating out the paraffin,

the oil and natural gas men use refrigeration to some extent in the salvage of gasoline. The chemical engineer, the baker, the candy maker, and the miner use refrigeration to secure

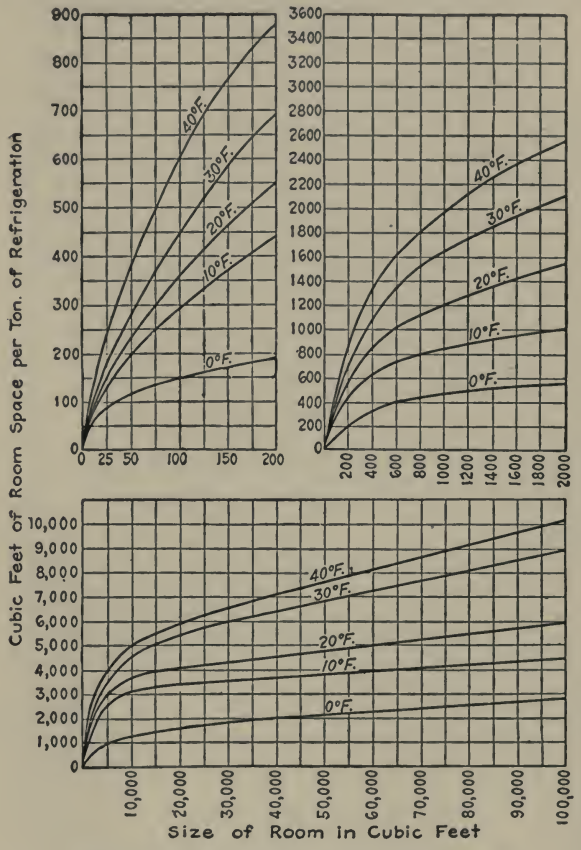


Fig. 108. Cubic feet of space per ton of refrigeration

standard conditions—a cold water, a cold air or a cold stratism of earth. As an example of the degree to which the industry is carried, it may be mentioned that the Haber process for

the fixation of atmospheric nitrogen is to form ammonia in a mixture of hydrogen and nitrogen by means of a special catalyst, and then to freeze the mixture to —20 deg. F. or lower until the ammonia is liquefied and separation may be secured by means of gravity and a modified trap.

In making an analysis of the different kinds of applications of refrigeration, one will be surprised by their similarity. As just noted, it is nearly always the matter of the cooling of a fluid, a liquid or a gas, which in turn does the work desired of it. However, there are a few other special applications which, because of their general use, are worthy of special mention. A few of these will be taken up in turn.

The Milk Industry

The milk industry is in an important one. Milk is a daily necessity, a food that cannot be dispensed with. In consequence of the distance from the source the city milk supply is subject to such delay that by the time it is ready to be served the milk may be 36 to 48 hours old. The result is that raw milk is frequently badly contaminated with bacilli of intestinal or lung variety, especially if the milk is allowed to remain even for a short time at a temperature of 60 or more deg. F. To reduce the danger from disease the pasteurizing process has been perfected. This consists of either bringing up the milk temperature to 140 to 150 deg. F. and holding at that temperature for thirty minutes (the preferred method) or of raising the temperature to 160 to 165 deg. F. and maintaining that temperature for only one-half a minute. After the pasteurizing process the temperature should be reduced to 35 or 45 deg. F. as quickly as practicable. This cooling may be accomplished in either of two ways—first, by means of a regenerative apparatus, and, second, by means of the old style Baudalot cooler.

Assume that the milk leaves the pasteurizer at about 140 deg. F. It would be sent to the regenerator, as shown

in one design in Fig. 110. The regenerator is a sort of counterflow exchanger. The raw milk is relatively cold and must be heated to 140 deg. F. and the pasteurized milk is at 140 deg. F. and must be cooled. By using a counter current device both of these results may be accomplished easily and cheaply. The last portion of the work of cooling is by means of brine, reduced in temperature by some method usual in mechanical refrigeration.

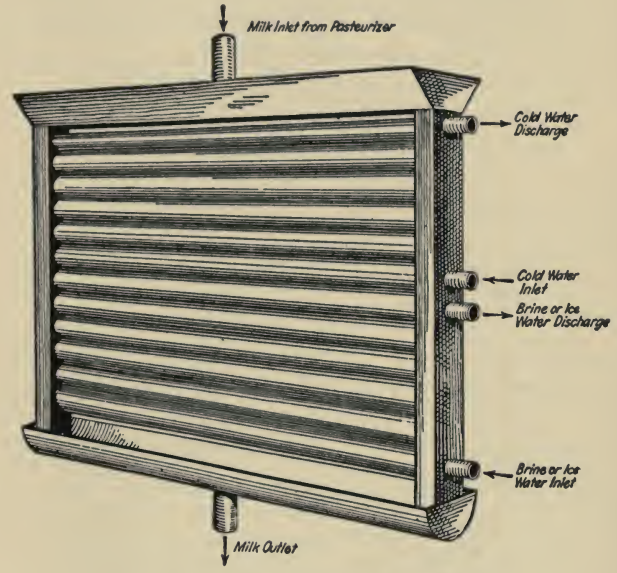


Fig. 109. The Bandalot milk cooler

The second method is to pipe the pasteurized milk directly into the cooler. This cooler is usually made of block tin (for cleanliness) and is arranged so that the milk flows on the outside. The refrigerating problem is a simple one, requiring brine of about 20 to 25 deg. F. The most important point is that the milk cooling after pasteurizing makes a heavy load, usually for a short part of the day only. There-

fore it usually works out best to have a brine storage tank of sufficient size to carry a large part of the load. In so doing the compressor may operate over a longer part of the

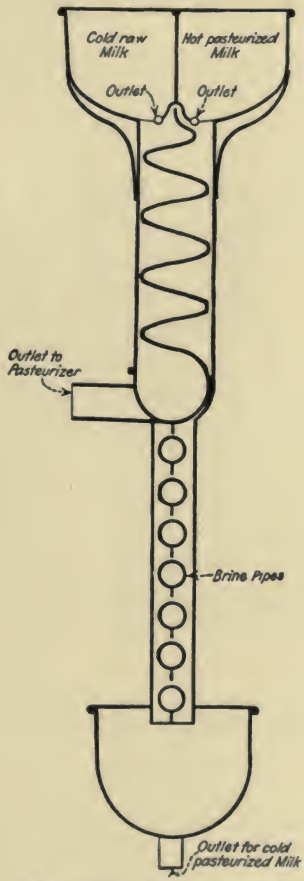


Fig. 110. An improved type of milk cooler

day, and the machine may be a smaller one than would be required otherwise. The amount of the refrigeration stored in the brine depends on the temperature rise of the brine.

Assuming this to be 20 deg. F. the brine storage necessary to cool 1,000 gals. of milk from 80 deg. to 40 deg. F. would be:

1000	×	8.58	×	40	×	0.9	=	Vol.	×	62.5	×	1.2	×	0.7	×	20
No. gals.	×	lb. per Gal.	×	temp. range	×	specific heat			×	weight of one cu. ft. of water.	×	B.t.u. capacity per unit vol. of brine.				

taking the specific heat and the specific gravity of the brine as 0.7 and 1.2 respectively. Solving, in cubic feet, the answer is

Volume = 294 cu. ft.

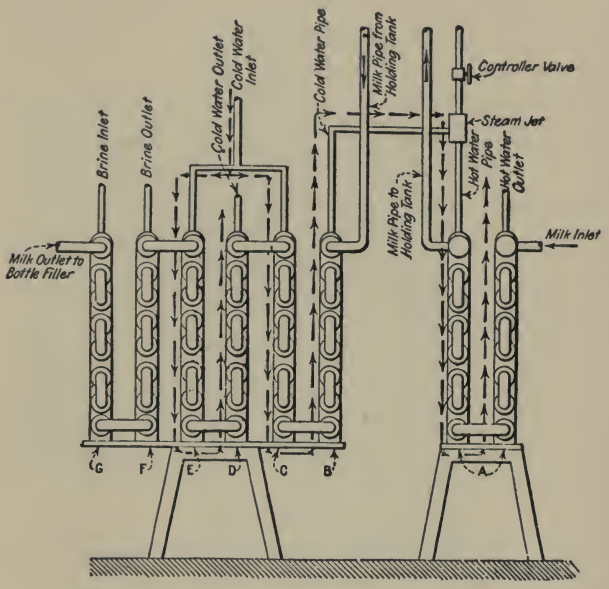


Fig. 111. Another type of milk cooler

Ice Cream Making. In ice making there is required frequently from 1.4 to 1.7 tons of refrigeration for 24 hours to produce a ton of ice. In other words, much more refrigeration has to be put into the manufacture of ice than it is possible to secure in the cooling effect by the melting of the ice after it is made. The reason for this is that in ice making the water has to be cooled first to 32 deg. F., then frozen,

and finally the ice cooled down to somewhere near the brine temperature. All this requires (assuming 70 degree water)

$$(70 - 32) + 144 + \frac{32 - 14}{2} = 191 \text{ B.t.u. per pound of ice.}$$

To this 191 B.t.u. should be added the losses by radiation, conduction, etc., and the meltage and other losses in removing the ice from the cans and storing the cakes in the storage rooms. The total input in heat units per pound of net ice is therefore from 210 to 230 B.t.u. from which only 144 B.t.u. is usually available for refrigerating purposes. In ice making, therefore, it is necessary to cool the water, freeze the ice, and finally cool the ice.

Ice cream making is practically the same kind of refrigerating work. About the only difference between ice making and ice cream making is in the details of the operation. Ice cream, of course, is not frozen in a solid mold like can ice, but as a rather soft mush, after which it is placed in a container and allowed to cool into a solid condition in the so-called hardening room. The refrigerating problem in ice cream making is in cooling the cream, freezing it to a plastic condition, and finally hardening it, a process which is much like the cooling of can ice down to the temperature of the brine in the ice tank.

Calculations of the duty required are not as positive as in ice making because the latent heat of fusion of the mixture of cream and the specific heat before and after freezing are not constant for all kinds of ice cream or ices. In general the mixture is usually placed in the freezer at about 50 deg. F. and the whipping of the cream occurs at about 26 deg. F. During the process of freezing the cream "swells" an amount of from 50 to 100 per cent. Using a latent heat for ice cream of 84 B.t.u. and the specific heat before and after freezing as 0.68 and 0.38 respectively, the refrigeration necessary to

make 100 gals. of cream from a mixture at 50 deg. F. to a final temperature of 0 deg. F. would be: $\frac{100}{1.8} \times 8\frac{1}{3} \times 1.0 =$ 463 lbs. (the weight of the cream to be frozen.)*

$(463 \times 0.68 \times 50—26) + (463 \times 84) + (463 \times 0.38 \times 26—0) =$
wt. spec. heat temp. range wt. latent. heat wt. spec. heat temp. range
$= 51{,}030$ B.t.u.

If, for example, the work had to be performed in one hour, there would be required $51{,}030 \div 12{,}000 = 4.25$ tons capacity. Of course there are the usual losses to be taken into account as well as the ice required for icing the deliveries and other incidentals still to be considered.

Air Conditioning and Dehumidifying

In theatres, packing houses, blast furnaces and in many other different installations the air is refrigerated and the humidity is regulated in some manner or other. In the packing house chill room, blood room, etc., the method used (as already mentioned) for refrigeration must quickly remove the animal heat and leakage losses, and also be able to remove the steam that is thrown off. The use of bunker and deck coils has been found to be less satisfactory than the *sodium chloride brine spray,* as will be described in detail later.

In auditorium work the amount of cooling of the air is small, perhaps only 10 or 15 deg. F., but frequently the problem is made more difficult by a desire to regulate the humidity also. In cooling air (except in especially dry climates as Arizona, New Mexico, Southern Idaho, etc.) not much drop of temperature is required before the dew point is reached. For ventilation it requires, then, that the relative humidity be regulated to between 70 to 75 per cent, and therefore in such work the actual cooling of the air must be carried to a point considerably below the temperature

* The weight of the cream depends on the swell and the specific gravity of the mixture. The above example is taken as an extreme case.

required (at which time the humidity will be 100 per cent) and then allowed to heat again.

In *blast furnaces,* where air conditioning is resorted to, the primary consideration is the regulation of the water content of the air so as to reduce the amount of coke consumed in the tower and to obtain a more regular or uniform product. In such work it is customary to precool the air to about 25 deg. F. where the water content is 20 grains per pound of dry air or 1.5 grains per cubic foot of free air.

In packing house work the humidity must be kept down, preferably to a point where the "fog" brought about by the hot carcasses will be made to disappear almost entirely. In all of these cases the method now used with best success is to employ cold water (for nominal air temperatures) and brine (for temperatures of the air below about 35 to 38 deg. F.) in the form of a spray through which the air is made to pass,—being washed, cooled and relieved of surplus moisture in the one process, very much after the method used in washing air for transformer or other electric machinery.

In all air refrigeration the problem presenting itself is about the same; first there is the cooling of the air, then condensation of the moisture, the cooling and then the freezing of the condensate and finally the cooling of the ice. In the use of the brine spray the condensate mixes with the brine and dilutes it. Then the weaker solution is drained off into the brine storage (where more salt is added or the concentration may be increased by evaporation). No freezing of the condensate occurs in this latter case. In figuring the heat to be removed under these circumstances it is usually sufficiently accurate to consider only the heat removed in condensing the moisture, and cooling the air and the condensate down to the temperature of the brine. All of these data may be obtained easily by means of the Carrier psychrometric chart (Fig. 112), as shown in the following example:

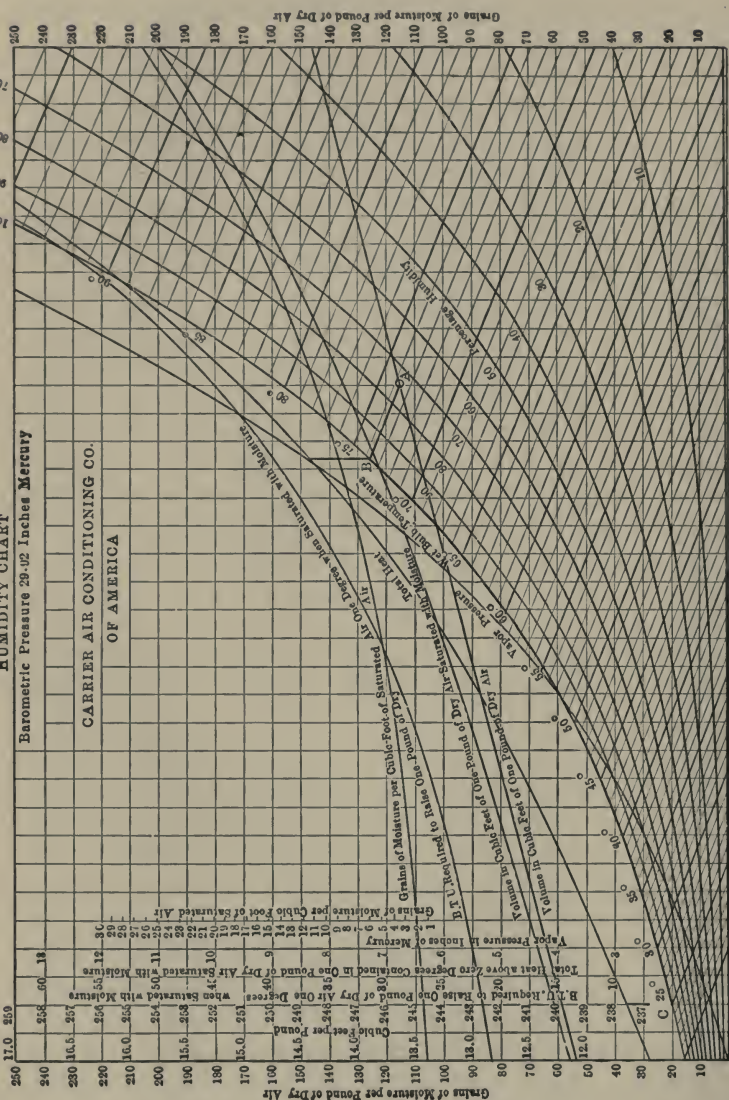
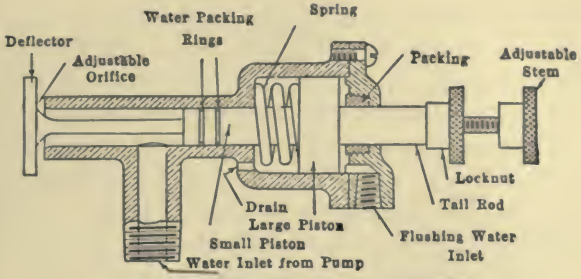
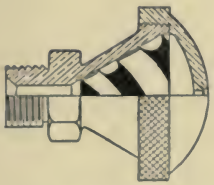

HUMIDITY CHART
Barometric Pressure 29.92 Inches Mercury
CARRIER AIR CONDITIONING CO.
OF AMERICA
Grains of Moisture per Pound of Dry Air
Dry Bulb Temperature
Wet Bulb Temperature
Relative Humidity
Vapor Pressure
Total Heat above Zero Degrees Contained in One Pound of Dry Air Saturated with Moisture
B.T.U. Required to Raise One Pound of Dry Air One Degree
Volume in Cubic Feet of One Pound of Dry Air
Cubic Feet per Pound
Grains of Moisture per Cubic Foot of Saturated Air
Vapor Pressure in Inches of Mercury
Total Heat in B.T.U. Required to Raise One Pound of Dry Air Saturated with Moisture
Apparatus Dew Point
Actual Temperature
Room Dew Point
Grains of Moisture per Cubic Foot of Saturated Air
Volume in Cubic Feet of One Pound of Dry Air
Total Heat in B.T.U. Required to Raise One Pound of Dry Air Saturated with Moisture
Grains of Moisture per Pound of Dry Air
Dry Bulb Temperature

Fig. 112. Psychrometric chart [For Air Calculations]

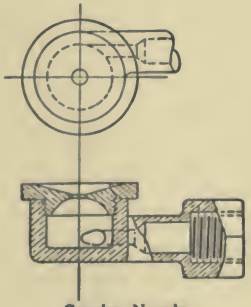


American Blower Co's Spray Nozzle



Sturtevant Atomizing Nozzle

Sturtevant Auxiliary Rain-Spray Nozzle



Carrier Nozzle

Webster Nozzle

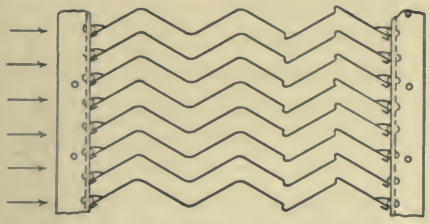
Fig. 113. Spray Nozzles for use of water or brine spray in air cooling and conditioning

Required the refrigeration necessary to cool 1,000 pounds of air from a temperature of 80 deg. F. and 75 per cent relative humidity to 25 deg. F. Locating this condition by

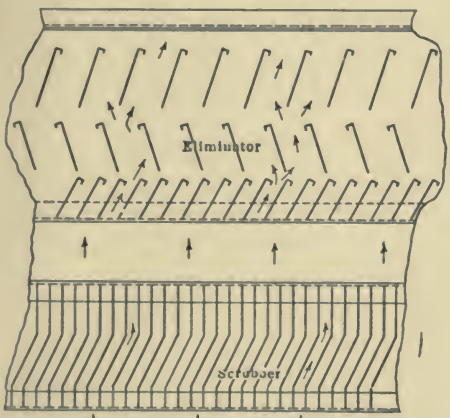
point A on the chart, the wet bulb temperature is seen to be B or 73.7 deg. F. and having a total heat (as shown by the diagram) of 36.5 B.t.u. Likewise the total heat corresponding to 25 deg. F. at C is 9.0 B.t.u. or a difference of 36.5—9.0= 27.5 B.t.u. per pound of dry air. The total refrigeration will be $1{,}000 \times 27.5 = 27{,}500$ B.t.u. The volume per pound may be found from the diagram also, for at 80 deg. a pound of dry air occupies 13.62 cu. ft. at standard atmospheric conditions and when saturated with moisture occupies 14.08 cu. ft., or a difference of 0.46 cu. ft. due to the water content. As our problem has 75 per cent moisture the volume of one pound of dry air and this amount of moisture is: $13.62 + 0.75 \times 0.46 = 13.97$ cu. ft. or 1000×13.97 or 13,970 cu. ft. as the volume of the air under consideration in the problem.

In cooling the air, as in the case of certain cold storage rooms, the main essentials are to get circulation and to bring all the air into contact with the spray. The nozzle is the all important detail, one that will provide a finely divided spray with a large angle and short trajectory being desired. Then it is necessary that there be sufficient space for the brine (or the water) to settle out by gravity or to be caught by the eliminators. Figs. 113 and 114 show details and methods used.

Water Cooling. Water cooling for hotels, office buildings, factories, etc., is best done by refrigeration, and circulated by means of some form of pump. As a rule water is best cooled by means of the Baudalot cooler (see Fig. 109), part of the coils being submerged, and the make-up and the return water from the system flowing through the distributing troughs and over the coils. The double pipe cooler is unsatisfactory because of the danger of freezing the water and bursting the pipe. The usual allowance for water cooling is 8 pints per person per 10 hour day. The problem of the calculation of the refrigeration required is the simplest of all our calculations, as it is the product of the number of

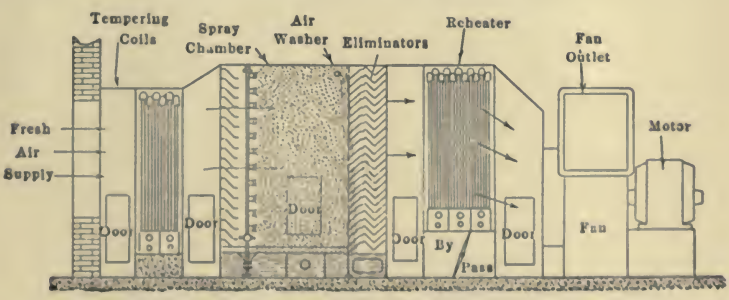


Relative Spacing, Proportions and Angle of Deflection of Carrier Eliminators



American Blower Co's Eliminator and Scrubber Plates

Old Design
Note Lack of Device for
Trapping and Separating Moisture



Sturtevant Heating, Ventilating and Air Conditioning System

Fig. 114. Water or brine eliminators for air cooling and conditioning

pounds of make-up water and the number of degrees of cooling of this water. In addition the circulated water usually increases from 2 to 3 degrees in temperature during the circuit, and this load must be neutralized by the refrigerating machine to prevent a temperature rise of the circulated water.

Conclusions. The steam and operating engineer will find nothing new in principle in studying the subject of refrigeration. The details are, of course, new, because of the cycle used, the kind of working medium employed, and the details of the operation and its many complications. In the fundamentals, however, there is usually some analogy between refrigeration and steam engineering, and one explanation makes both processes clear.

Although this material has been written primarily for the operating engineer for steam machinery it is hoped that others will find the explanations easily understandable. In this branch of engineering it may be said that when everything else is considered the fact remains that "heat cannot of itself run up hill," but must pass from a hotter to a colder condition; that the refrigerating machine is a "temperature pump" so that condensation of the refrigerant may occur at a nominal temperature in the condenser; and that (as in steam boilers, condensers and steam evaporators) refrigeration to a large extent is a matter of heat transfer.

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